



Proceedings

ASPE 2012 Annual Meeting

October 21-26, 2012

Hyatt Regency La Jolla at Aventine
San Diego, California

American Society for

Precision Engineering

P.O. Box 10826, Raleigh, NC 27605-0826
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Proceedings of:

The Twenty-Seventh Annual Meeting of

The American Society for

Precision Engineering

October 21 to October 26, 2012
Hyatt Regency La Jolla at Aventine
San Diego, California

The American Society for Precision Engineering (ASPE) is a multidisciplinary professional and technical society concerned with research and development, design, manufacture and measurement of high accuracy components and systems. ASPE activities encompass relevant aspects of mechanical, electronic, optical and production engineering, physics, chemistry, and computer and materials science. Membership is open to anyone interested in any aspect of precision engineering.

Founded in 1986, ASPE provides a new focus for a diverse but important community. Other professional organizations have covered aspects of precision engineering, always as a sideline to their principal goals. ASPE is based on the core of generic concepts necessary to achieve precision in any application; independent of discipline, ASPE intends to be the focus for precision technology – and to represent all facets from research to application.

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Preface

This book comprises the proceedings of the 27th ASPE Annual Meeting. The contributions reflect the authors' opinions and are published as presented to ASPE, without change. Their inclusion in this publication does not necessarily constitute endorsement by the ASPE, or its editorial staff.

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Welcome Note

Welcome to San Diego and the Twenty-Seventh Annual Meeting of The American Society for Precision Engineering. The success of the Society reflects the increasing importance of precision engineering in a wide variety of fields, from manufacturing to microelectronics to basic science. The Society serves as a focus for precision engineers across all of these fields. The Annual Meeting has evolved into a premier international forum for the exchange of ideas and presentation of research results relating to precision engineering, metrology, controls, and system integration. Precision engineers and scientists from private industry, government laboratories, and universities meet to learn about the latest developments and to exchange ideas about the future directions of these technologies.

The 2012 ASPE Annual Meeting continues the tradition of offering two full days of tutorials presented by some of the foremost precision engineers and scientists in the world. The technical sessions and poster presentations offer the latest research results in the areas of ultraprecision machining, metrology, controls, precision grinding, micro-positioning, materials processing, design, precision transducers, surface profilometry, and error compensation. The commercial exhibits will provide attendees with the opportunity to view and discuss the latest precision engineering equipment, products, and services.

The conference organizing committee is proud to present the program for the Twenty-Seventh Annual Meeting of the ASPE. We welcome your participation and your feedback on how to make future meetings even better.

2012 Organizing & Technical Program Committee

Senajith (Seno) Rekawa

Lawrence Berkeley National Laboratory
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Alex Sohn, North Carolina State University
Mark A. Stocker, Cranfield Precision, Division of Cinetic Landis Ltd.
Pradeep K. Subrahmanyam, KLA-Tencor
Hans-Jochen Trost, MicroFab Technologies, Inc.
Allen Y. Yi, The Ohio State University

2012 Scholarship Recipients

Maria J. Telleria, Massachusetts Institute of Technology
Christopher Tyler, University of North Carolina-Charlotte

2012 Scholarship Sponsors



2012 Annual Meeting

Contents

Preface	ii
Welcome Note	iii
Organizing Committee – 2012 Scholarship Recipients and Sponsors	iv
Table of Contents	v
Program.....	vi
Keynote Address	vii
Technical Exhibitors	viii
Open Forum: Panel Discussion	ix
Sustaining Corporate Sponsors	x
Corporate Sponsors	xi
Lifetime Achievement Award.....	xii
Technical Paper Index	1 - 11
Technical Papers	15 - 537
Author Index	539 - 543

Program

2012 ASPE Annual Meeting

Time	Sun., Oct. 21	Mon., Oct. 22	Tues., Oct. 23	Wed., Oct. 24	Thurs., Oct. 25	Fri., Oct. 26
8:00	Registration		Registration and Early Coffee			Enjoy San Diego
9:00	Tutorials	Tutorials	Technical Session I	Technical Session III	Technical Session VI	
10:00			Break	Break	Break	
11:00			Technical Session II	Technical Session IV	Technical Session VII	
12:00			Lunch and Committee Meetings	Awards Lunch and Business Meeting	Lunch and Open Forum	
1:00						
2:00	Tutorials	Tutorials	Commercial Session	Technical Session V	Technical Session VIII	
3:00			Midway Talk & Break	Break		
4:00			Poster Session I	Poster Session II		
5:00			Hospitality Hr.	Hospitality Hr.		
6:00		Welcome Reception	Board Bus for Dinner			
7:00		Keynote Address	Dinner at USS Midway			
8:00						
9:00						

Aventine Ballroom ABC

Keynote Address
Technical Sessions
Commercial Session
Midway Information Talk

Aventine Ballroom DEFG

Exhibits
Posters
Breaks
Hospitality Hours

Pavilion

Welcome Reception
Tuesday Lunch (Committee Meetings)
Wednesday Lunch (Business Meeting)
Thursday Lunch (Open Forum)

Foyers

Registration
Early Morning Coffee
Posters

Check hotel event board for exact location of tutorials and for location to board buses for the Midway Dinner.



Exhibit Open Hours

Keynote Address

Elie Homsy, P. E.

Flatiron Construction Corporation

Monday, October 22, 2012 7:00 p.m

Session chair: Senajith (Seno) Rekawa, Lawrence Berkeley National Laboratory



Elie Homsy is Executive Vice President of Engineering Services overseeing engineering and international operations at Flatiron. With a civil engineering degree from the University of Texas, he has over 30 years of experience in the construction industry.

Regarded as an industry leader, Homsy has developed several new and innovative solutions that have garnered numerous awards. He was granted several U.S. and international patents including the Tilting Lead Gantry, the Formless Precast Edge Girder, and the Crawler Mounted Tilting Lead, giving Flatiron a competitive edge in the bridge building market. Homsy was awarded numerous recognitions including the

ENR Top 25 Newsmakers, the ASBI Leadership Award, the Nova Innovation Award, and was recently elected to the Academy of Distinguished Alumni at the University of Texas.

Homsy was also a leader in forming Flatiron's partnership with Bridges to Prosperity, a volunteer-based charity that provides isolated communities with access to essential health care, education and economic opportunities by building footbridges.

Precision Engineering on an 800 Ton Scale

Geometry control in the precasting and erection of large bridge components is important in precast segmental bridge technology. Different factors affect the individual shape of each component as well as their impact on the final shape of the completed structure. It is precision engineering applied on a large scale to concrete components weighing 80 to 800 tons and assembled in 100 to 1600 foot-long spans.

Cooperating Sponsor

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Technical Exhibitors

4D Technology Corporation	New Way Air Bearings, Inc.
Aerotech, Inc.	Newport Corporation
Alicona USA	Nikon Corporation
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attocube Systems AG	Optical Perspectives Group LLC
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Machine Dynamics Research Lab. – Penn State University	Taylor Hobson (Division of AMETEK)
Makino	Trust Automation
Moore Nanotechnology Systems, LLC	Western Environmental Corporation
	Zygo Corporation

Commercial Session

Tuesday, October 23, 1:30 - 3:00 p.m.

Session Co-Chairs: Byron R. Knapp (Professional Instruments Company, Inc.)
and Mark T. Kosmowski (Interdyne Systems)

The Commercial Session will take place early in the week on Tuesday afternoon. This is a special session where Exhibitor representatives are invited to make brief presentations on products associated with precision engineering. This provides participants with the opportunity to receive timely information on technologies that have been commercialized into products and services, as well as information on advances that can be expected in the near future.

Open Forum

Thursday, October 25, 2012 12:00 noon - 1:30 p.m.

Session Chair and Moderator: Kate M. Medicus (Optimax Systems, Inc.)

Panel Discussion

Wide Spread Dissemination of Precision Engineering Principles

Moderator led, this panel, along with the audience, will discuss how the ASPE community can better disseminate Precision Engineering principles. This is a follow up from a discussion by Tyler Estler at the 2010 ASPE Annual Meeting regarding machine manufacturers not using the correct vocabulary. The goal of the panel is to determine concrete steps that ASPE members can do to spread the Precision Engineering message.

Some questions for the panel are as follows: Should ASPE be teaching classes or hosting seminars at other societies' meetings? Do we need to focus on internal education before we look externally? Is this topic too big? Is this dissemination the job of Universities, National Labs, or ASPE?

Questions will be taken from the moderator and the audience. Please come ready with questions or topics for discussion. If you prefer not to voice your question during the session, please speak to the moderator before the session.

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2012 ASPE Lifetime Achievement Award



Sir David McMurtry

As inventor of the touch trigger probe and co-founder of Renishaw plc, Sir David McMurtry's contributions to the field of precision metrology and manufacturing are numerous and global in nature. Sir David's electronic touch trigger probe transformed the field of contact metrology producing dramatic improvements in measurement precision and accuracy. His technical and entrepreneurial leadership at Renishaw has spanned almost four decades, resulted in over 150 Renishaw patents, and provided manufacturers across multiple industries the tools necessary to measure and produce products traceable to international standards.

Sir David McMurtry, CBE, RDI, FRS, FEng, CEng, FIMechE, is Chairman and Chief Executive of Renishaw plc. with responsibility for group technology. Renishaw was founded in 1973 by Sir David and John Deer (now Deputy Chairman of Renishaw plc), a colleague from Rolls-Royce, to carry on the business of producing and selling the touch-trigger probe he invented and designed in the early 1970s, while employed by the Rolls-Royce group at its Aero-engine Division at Filton, Bristol.

Sir David was awarded Royal Designer for Industry (RDI) and was made a Commander of the Order of the British Empire (CBE) "for services to Science and Technology." Sir David is a visiting Professor at the University of Huddersfield and was awarded an honorary Doctorate of Engineering from the University of Birmingham. He has also been awarded honorary degrees of Doctor of Engineering from Heriot-Watt University, University of Bristol and the University of Bath. Sir David is a Fellow of the Society of Manufacturing Engineers (USA), a Fellow of the Institution of Mechanical Engineers, a Fellow of the Royal Academy of Engineering (FEng) and a Fellow of the Royal Society. He has received many awards including the 7th ND Marketing Award in Japan; the Champions of Metrology Award from the Metrology for World Class Manufacturing; the Duncan Davies Memorial Medal by the Research and Development Society; Master Entrepreneur, South Region in the Ernst & Young LLP "Entrepreneur of the Year" award; Lifetime Achievement Award at the annual West of England Business Awards; Master of Manufacturing from the official magazine of the Society of Manufacturing Engineers; the General Pierre Nicolau Award from the International Academy for Production Engineering and the Swan Medal of The Institute of Physics.

Technical and Poster Sessions Index

The technical program of the 2012 ASPE Annual Meeting contains 138 papers on precision engineering advances. Papers that lent themselves to significant verbal interaction, concise but important discoveries, or strongly visual or tactile subjects have been selected for the poster presentation. Authors will be in attendance to discuss their work during the Poster Sessions on Tuesday, October 23 and Wednesday, October 24, from 3:30 to 5:00 p.m.

Session I

Progress in Precision

Tuesday, October 23, 2012, 8:30 AM - 10:00 AM

Session Chair: Thomas A. Dow (N.C. State University), Eric S. Buice (Lawrence Livermore National Laboratory) and Brian P. O'Connor (Aerotech, Inc.)

- 1. Calibration of Areal Surface Topography Measuring Instruments: Are We There Yet?**
Leach, R. K.; Giusca, C. (National Physical Laboratory) and Coupland, J. M.;
Mandal, R. (Loughborough University) and Foreman, M. (Imperial College London) 15
- 2. Production of Large Optics Supporting Experimental Campaigns at the National Ignition Facility**
***Invited Paper**
Taylor, J. S. (Lawrence Livermore National Laboratory)
- 3. Computed Tomography in Metrology**
Bauza, M. B. (Carl Zeiss Industrial Metrology LLC) and Lettenbauer, H. (Carl
Zeiss Industrielle Messtechnik GmbH). 20

Session II

Design of Precision Machines

Tuesday, October 23, 2012, 10:30 AM - 12:00 PM

Session Chair: Mark A. Stocker (Cranfield Precision, Division of Cinetic Landis Ltd.) and Mark T. Kowmowski (Interdyne Systems)

- 1. Synthesizing Precision Flexures That Decouple Displacement-Based Actuators**
Hopkins, J. B. (Lawrence Livermore National Laboratory) and
McCalib Jr., D. B. (Oregon Institute of Technology) 26
- 2. Water-Cooled Heat Load Chopper for High-Energy Synchrotron Experiments**
Knapp, B. R.; Oss, D. D.; Arneson, D. A (Professional Instruments Company, Inc.) and Graber, T.;
Henning, R. (The University of Chicago) and Marsh, E. (The Pennsylvania State University) 30
- 3. Air Bearing for Next Generation CT Scanners**
Soetens, J. B. M. (Philips Innovation Services) and Soemers, H. M. J. R. (Philips
Innovation Services and Twente University). 34
- 4. Optimising Machine Performance by Novel Design**
Miller, G. J. (Cranfield Precision, Division of Cinetic Landis Ltd.) 38
- 5. Viscous Shear in Air Bearing Gaps for Precise Web Tension and Temperature Control**
Devitt, A. J.; Allen, J. (New Way Air Bearings). 42

**No abstract available*

Session III

Precision Manufacturing

Wednesday, October 24, 2012, 8:30 AM - 10:00 AM

Session Chair: Bradley H. Jared (Sandia National Laboratories) and Hans-Jochen Trost (MicroFab Technologies, Inc.)

- 1. Improvement of Heat Transfer Coefficient of Boiling Surface on Nanostructured Metal Using Nanoimprinting**
Watanabe, Y.; Nagato, K.; Shikazono, N.; Shimamoto, K.; Nakao, M. (The University of Tokyo) 46
- 2. Remaining Useful Tool Life Predictions Using Bayesian Inference**
Karandikar, J. M.; Schmitz, T. L. (University of North Carolina at Charlotte) and Abbas, A. E. (University of Illinois at Urbana-Champaign) 50
- 3. Mechanisms of CMP Pressure Profile Generation**
Panas, R. M.; Nyovanie, P.; Du, L.; Culpepper, M. L. (Massachusetts Institute of Technology) 54
- 4. An Analysis of Polishing Forces in Magnetic Field Assisted Finishing**
Ganguly, V.; Schmitz, T. L. (University of North Carolina at Charlotte) and Graziano, A.; Yamaguchi, H. (University of Florida) 58
- 5. Integrated Microfluidic Device for Forensic Fiber Dye Analysis**
Sohn, A.; Garrard, K. P.; Dow, T. A.; Robichaud, G.; (North Carolina State University). 62

Session IV

Interferometry and Optics

Wednesday, October 24, 2012, 10:30 AM - 12:00 PM

Session Chair: Kate M. Medicus (Optimax Systems, Inc.) and Robert D. Grejda (Corning Tropel Corporation)

- 1. High-Precision Absolute Thickness Measurements of Optically Thick Components**
Greenfield, N.; Reibel, R.; Kozicki, D. E.; Roos, P.; Berg, T (Bridger Photonics, Inc.) 66
- 2. Interferometric-Tactile Hybrid-Stylus for Precision Form, Roughness and Diameter Measuring Machines**
Drabarek, P.; Fleischer, M. (Robert Bosch GmbH). 70
- 3. Seismic Platform Interferometer - An Auxiliary Sensor for Use in the Advanced LIGO**
Clark, D. E.; Lantz, B. T.; DeBra, D. B. (Stanford University) 73
- 4. Miniature, Fiber-Coupled 3-DOF Interferometer for Precision Micro-Motion Stage Metrology**
Gillmer, S.; Smith, R. C. G.; Ellis, J. D. (University of Rochester) and Woody, S. C.; Tarbutton, J.; (InsituTec, Inc.) 77

Session V

Precision Metrology

Wednesday, October 24, 2012, 1:30 PM - 3:00 PM

Session Chair: Marcin B. Bauza (Carl Zeiss Industrial Metrology) and Jonathan D. Ellis (University of Rochester)

- 1. A Novel 6-Axis Position Sensor**
Rao, N.; Lu, X. (The University of British Columbia) 81

2. Trinano N100 3D Measurements with Nanometer Repeatability and Effects of Probe-Surface Interaction Bos, E.; Moers, T.; van Riel, M. (Xpress Precision Engineering B.V.)	85
3. Precision Mass Property Measurements Using a Five-Wire Torsion Pendulum Swank, A. J. (NASA Glenn Research Center)	89
4. Error Budgeting as Applied to the Design of the International X-Ray Observatory Mandrel Measuring Machine Thompson, D. C.; Youden, D. H. (Praecis, Inc.)	93

Session VI

Nano-Positioning

Thursday, October 25, 2012, 8:30 AM - 10:00 AM

Session Chairs: Deming Shu (Argonne National Laboratory) and Richard K. Leach (National Physical Laboratory)

1. Planar Magnetic 6D Levitating System with Compact Sensor Schaeffel, C.; Katzschmann, M.; Mohr, H.-U. (Institut für Mikroelektronik-und Mechatronik-Systeme gemeinnuetzige GmbH) and Glöss, R.; Rudolf, C.; Mock, C. D.; Walenda, C.; Liang, R. (Physik Instrumente (PI) GmbH & Co.KG)	99
2. Positioning Performance of a Planar Nanopositioning System with 100 mm Circular Travel Range Hesse, S.; Katzschmann, M.; Schaeffel, C. (Institut für Mikroelektronik - und Mechatronik - Systeme gemeinnützige GmbH) and Buechner, H.-J. (Technische Universität Ilmenau)	103
3. Ultrasonic Indenting of Nanofeatures Nowak, J.; Dow, T. A.; Scattergood, R. O. (North Carolina State University)	107
4. Plasmonic Nano-Lithography Machine (PNLM) Saad, M. E.-S.; Hocken, R. J. (University of North Carolina at Charlotte)	111
5. Progress in the Development of Precision Positioning Stages for X-Ray Scanning Nanoprobes Shu, D.; Maser, J.; Lai, B.; Liu, W. (Argonne National Laboratory;) and Yan, H.; Nazaretski, E.; Lauer, K.; Kim, J.; Chu, Yong S. (Brookhaven National Laboratory)	116

Session VII

Precision Machining Processes

Thursday, October 25, 2012, 10:30 AM - 12:00 PM

Session Chair: Byron R. Knapp (Professional Instruments Company, Inc.) and Pradeep K. Subrahmanyam (KLA-Tencor)

1. Process Damping Analytical Stability Analysis and Validation Tyler, C.; Schmitz, T. L. (University of North Carolina at Charlotte)	120
2. Nanocoining Sub-Micrometer Features Zdanowicz, E. M.; Dow, T. A.; Scattergood, R. O. (North Carolina State University)	124
3. Ellipse Shape Effect on Tool Wear in Elliptical Vibration-Assisted Machining Lane, B. M. (National Institute of Standards & Technology) and Dow, T. A. (North Carolina State University)	128
4. Cutting Temperature and Vibration Control in High-Performance Machining of Titanium Alloys Wang, Z.; Larson, M. W.; Garrison, S. J. (Makino, Inc.)	132

5. Study on Curved Line Cutting Using Flexible Circular Saw

Yamada, Y.; Sasahara, H. (University of Agriculture & Technology) and Osumi, N. (Orion Tool Co., Ltd.) and Takasugi, A. (Shoda Iron Works Co., Ltd.) 136

Session VIII

Control of Precision Machines

Thursday, October 25, 2012, 1:30 PM - 3:00 PM

Session Chair: Luis A. Aguirre (3M Company) and Stephen J. Ludwick (Aerotech, Inc.)

1. Real-Time Feed Optimization for Spline Toolpaths Employing Uninterrupted Acceleration Profiling

Erkorkmaz, K. (University of Waterloo) 140

2. Driving Mode Control of A Multilayered Thin Electrostatic Actuator Supported by Lubricating Oil for Ultra-Precision Positioning

Sato, K.; Ghazaly, M. M. (Tokyo Institute of Technology) 144

3. Reduction of the Residual Vibrations of Ultra-Precision Manufacturing Machines via Mode Coupling

Okwudire, C. E. (University of Michigan) and Kim, Chang-Ju; Kim, Jongsoo (Digital Technology Laboratory Corporation) 148

4. Control Schemes for Microfluidic Viral DNA/RNA Amplification

Pak, N.; Holst, G. L.; Phaneuf, C. R.; Saunders, D. C.; Forest, C. R. (Georgia Institute of Technology) 152

5. A New Hybrid Control Strategy Incorporating Fast-Tool Servos

Wilson, C. S. (Delta Tau Data Systems, Inc.) 156

Poster Session I

Tuesday, October 23, 2012, 3:30 PM - 5:00 PM

Control of Precision Machines

1. Wide Bandwidth Control of Ballscrew Drive Using Center of Mass Location Feedback

Hosseinkhani, Y.; Erkorkmaz, K. (University of Waterloo) 160

2. Recent Advancements in Motion Control with Long-Term Nanoscale Stability, Including a Novel, Comparative Validation Technique

Jordan, S. C. (PI (Physik Instrumente) L.P.) 164

3. Rotation Control and Disturbance Force Compensation of Air Turbine Spindle for Precision Machining Using High-Precision, Quick-Response Pneumatic Pressure Regulator

Kato, T.; Higashijima, G. (Fukuoka Institute of Technology) and Yazawa, T.; Otsubo, T.; Nozaki, Y. (Nagasaki University) and Tanaka, K. (Toshiba Machine Co., Ltd.) 169

4. Analysis of the Effect of PWM on Position Control Using Dynamic Error Budgeting

Niglas, P.; Erkorkmaz, K.; Ismail, F. (University of Waterloo) 173

5. Model Based Calculation of the Tool-Path Smoothing Tolerances

Sellmann, F.; Weikert, S. (inspire AG) and Wegener, K. (Inst. of Machine Tools and Manufacturing) 177

Design of Precision Machines

1. Retrofitting a Parker Liberty One Cylindrical Grinder with Air Bearing Spindles Arneson, D A. (Professional Instruments Company, Inc.)	180
2. Identification of the Influence of the Different Air Bearings Comprised in an Axis of Rotation System on the Radial Error Motion Cappa, S.; Reynaerts, D.; Al-Bender, F. (Katholieke Universiteit Leuven)	182
3. Design of a Precision Flexure-Based Vibration Microtome for Whole Organ Imaging Chen, S.-C. (The Chinese University of Hong Kong) and Panas, R. M.; Culpepper, M. L. (Massachusetts Institute of Technology) and Ragan, T. (TissueVision, Inc.).	186
4. Uncertainty Analysis of Geometric Error of a Super-Size Floor Type Boring Machine Cho, H.-G.; Min, B.-K.; Chung S.-C. (Hanyang University).	190
5. Design & Simulation of A Precision Rotating Valve System for Neutron Imaging Fitsos, P.; Edson, S. L.; Hall, J.; Peaslee, P.; Rusnak, B. (Lawrence Livermore National Laboratory).	194
6. Mechanism of Driving Force Generation in Two-Dimensional Non-contact Transfer Method by Near-Field Ultrasonic Levitation Furutani, K.; Ono, M. (Toyota Technological Institute).	198
7. Surface Damage Growth Mitigation on KDP/DKDP Optics Using Single-Crystal Diamond Micro-Machining Ball End Mill Contouring Geraghty, P.; Thigpen, R. (Lawrence Livermore National Laboratory).	202
8. Ball Bearing Analyzer Apparatus Keller, E. E.; Marsh, E. R.; Deakyne, T. R. (The Pennsylvania State University) and Knapp, B. R.; Arneson, D. A. (Professional Instruments Company, Inc.)	206
9. Experimental Apparatus for Metrology of Partial Cylinders Keller, E. E.; Marsh, E. R.; Deakyne, T. R. (The Pennsylvania State University) and Vallance, R. R. (nanoPrecision Products, Inc.) and Knapp, B. R. (Professional Instruments Company, Inc.).	209
10. Prediction and Experimental Verification of Motion Errors in Porous Air Bearing Stages Khim, G.; Oh, J.-S.; Park, C.-H. (Korea Institute of Machinery & Materials (KIMM))	213
11. A Spindle for a Large Liquid Mirror Telescope Liebers, M. J.; Arneson, D. A.; Sanner, S. L. (Professional Instruments Company, Inc.)	217
12. Geometric Error Modeling of A Super-Size Floor Type Boring Machine Min, B.-K.; Shin, D.-S.; Chung S.-C. (Hanyang University).	220
13. Development of Precision Micromachining Center and Applications Technologies Mochida, H.; Yamaguchi, T. (Makino Milling Machine Co., Ltd.) and Bradford, J. W. (Makino, Inc.)	224
14. Characterization of SI-Traceable Vacuum Compatible Precision Nanoindentation Platform Nowakowski B. K.; Smith, D. T.; (National Institute of Standards & Technology) and Smith, S. T. (University of North Carolina at Charlotte)	228
15. Fabrication of a Green Caterpillar Type In-Pipe Mobile Robot Driven by Pneumatic Bellows Actuators Ono, M.; Sakamoto, M. (Tokyo Metropolitan College of Industrial Technology)	232

16. Characterization of Vertical and Pitch Stiffness of Porous Vacuum Preloaded Air Bearings Ruszkowski, A.; Erkorkmaz, K.; Stubbley, G. (University of Waterloo)	236
17. Development of Simple Biaxial-Tensile-Testing Apparatus Using Lathe Chuck Sakamoto, M.; Ono, M.; Hiroi, T. (Tokyo Metropolitan College of Industrial Technology)	240
18. Kinematic Solar Dish - Initial Results Shelef, B.; Erez, S. (Heliocentric Solar Technologies)	244
19. Non-Contact Position Control via Fluid Shear Force Sung, E.; Fucetola, J. J.; Schattenburg, M. L.; Heilmann, R. K. (Massachusetts Institute of Technology) and Devitt, A. J. (New Way Air Bearings)	245
20. System Engineering to Meet Contradicting Requirements van der Straaten, G.; van Rens, P.; Peijnenburg, A. T. (VDL Enabling Technologies Group)	249
21. Silicon Carbide Joints for Thermo-Mechanically Stable Assemblies Wilson, M.; Fellows, J.; Anderson, H.; Lewinsohn, C. (Ceramtec, Inc.) and Cavanaugh, D. (CoorsTek Technical Ceramics)	252

Interferometry and Optics

1. Dual Moiré and Laser Interferometry as a Metrology Tool Abolbashari, M.; Gerges, A. S.; Davies, A. D.; Farahi, F. (University of North Carolina-Charlotte)	256
2. Deformation Error of Epoxy-Replicated X-Ray Mirror Chon, K. S. (Catholic University of Daegu) and Namba, Y. (Chubu University)	260
3. Experimental Evaluation of Phase Digitizing Nonlinearity Correction in Heterodyne Interferometry Ganguly, V.; Schmitz, T. L. (University of North Carolina at Charlotte) and White, D.; Yun, J. L. (Agilent Technologies, Inc.)	264
4. Study on a 1D Homodyne Encoder and a Novel 2D Homodyne Encoder Guan, J.; Köchert, P.; Weichert, C. (Physikalisch Technische Bundesanstalt (PTB)) and Tutsch, R. (TU Braunschweig)	268
5. An Automatic Focusing Module for White Light Scanning Interferometer Kim, T.; Lee, M.-G.; Jeon, Y.; Yeom, H. H.; Choi, B. J. (Ajou University) and Jeong, J. (Korea University) and Yoo, H. (Hanyang University).	272
6. Multi-Point Trilateration: A Novel Approach For Distributed Metrology Reibel, R. R.; Roos, P. A.; Greenfield, N.; Wilson, C. R.; Kozicki, D. E. (Bridger Photonics, Inc.)	276
7. Sensitivity of Optical Fibers to Mechanical and Thermal Perturbations for Interferometric Applications Smith, R. C. G.; Wang, C.; Ellis, J. D. (University of Rochester)	280
8. Optomechanical Characterization of Large Wafer Stepper-Optics with Respect to Centering Errors, Lens Distances and Center Thicknesses Stickler, D.; Lüerß, B.; Langehanenberg, P.; Heinisch, J.; Dumitrescu, E. (TRIOPTICS GmbH)	284

Nano-Positioning

1. Precision Flexure Stage Utilizing Parasitic Motion Burnham-Fay, E.; Smith, R. C. G.; Gillmer, S.; Ellis, J. D. (University of Rochester)	288
2. Patterning Phase-Coherent Lines on Large Substrates using Direct-Write Lithography Fucetola, J. J.; Heilmann, R. K.; Schattenburg, M. L.; Sung, E. (Massachusetts Institute of Technology)	292

Precision Manufacturing

1. Accurate Fabrication by Improvement of Lamination Path on Direct Metal Lamination Using Arc Discharge Abe, T.; Katagiri, M.; Sasahara, H. (Tokyo University of Agriculture and Technology)	296
2. Machining and Grinding Critical Components for the Mars Rover, Curiosity Arneson, D. A. (Professional Instruments Company, Inc.) and Hanson, B (Andrew Tool & Machining) and Sheridan, T. (Lorien Consultants Group, Inc.)	300
3. Metrological Solutions for Detective Optics Manufacture de la Maza, B. (Innovalia Metrology) and Ventura, T.; Pont, P. C. (Datapixel)	303
4. Temperature Measurement and Study of the Thermal Effects During the Micro-Laser Assisted Machining Process Del Orbe, D. V.; McKinley, J. D.; Ghantasala, M.; Patten, J. A.; Ravindra, D. (Western Michigan University).	304
5. Superfinishing Performance of Bearing Steel with CeO₂ Superabrasive Stone Furushiro, N.; Yamaguchi, T. (Kansai University) and Matsumori, N.; Ogura, H. (Mizuho Co., Ltd.) and Shimada, S. (Osaka Electro-Communication University)	308
6. Method for Measuring Thermal Distortion of Large Machine Tools Gomez-Acedo, E.; Olarra, A.; Zubieta, M. Kortaberria, G. (IK4-Tekniker).	312
7. Orientation Control of Anisotropic Particle by Shear Force Ikeshima, S.; Nagato, K.; Yamauchi, T.; Hamaguchi, T.; Nakao, M. (The University of Tokyo).	316
8. Design of Ignition Tuning Targets for Indirect Drive Ignition Experiments on NIF Kroll, J. J.; Bhandarkar, S.; Edson, S. L.; Gonzales, S.; Horner, J. B.; Dzenitis, E. G.; Taylor, J. S.; Nathan, B. (Lawrence Livermore National Laboratory) and Barker, D.; Vonhof, S.; Howland, F. A.; Heidl, B. F. (Akimia Infrastructure Services, Inc.) and Alger, E. T. (General Atomics)	320
9. A Shop Air Supply for a Hundred Spindles Liebers, M. J.; Arneson, D. A.; Sanner, S. L. (Professional Instruments Company, Inc.)	321
10. Automatic Centering of Precision Turned Parts on a Vacuum Chuck Marston, Z. C.; Dow, T. A.; Garrard, K. P.; Sohn, A. (North Carolina State University)	325
11. The Design, Fabrication, and Assembly of Material Equation-of-State Targets for the National Ignition Facility Nederbragt, W. W.; Lord, D.; Peterson, S. C.; Swift, D. W.; Wilson, M. J.; Bettencourt, K.; Mirkarimi, P.; Felker, S.; Kumar, C. (Lawrence Livermore National Laboratory)	329
12. Novel Approach for Slicing of Silicon Carbide by Multi-wire EDM using Wire Electrode with Track-shaped Section Okamoto, Y.; Kimura, A.; Okada, A. (Okayama University) and Ohya, J. (Toyo Advanced Technologies Co., Ltd.) and Yamauchi, T. (Tokusen Kogyo Co., Ltd.)	331
13. Digital Lithography Masking Method on Fabrication of Presice Uniform Micro-Feature Array with 16 μm Resolution Saragih, A. S.; Ko, T. J. (Yeungnam University)	335
14. Machine Tool Genome Project Case Study Schmitz, T. L.; Dutterer, B. S. (University of North Carolina at Charlotte)	339
15. Enhancement of Power Efficiency of Solid Oxide Fuel Cells by Aligning Microstructure Paths of Ion/Electron/Gas in Anode Shinagawa, S.; Nagato, K.; Shikazono, N.; Nishibashi, K.; Hamaguchi, T.; Nakao, M. (The University of Tokyo)	343
16. Adhesion Control of Coating Film by Texturing Watanabe, A.; Mizoguchi, T.; Kobayashi, Y.; Shirai, K. (Nihon University)	347

17. Ultraprecision Engineering Approach in the Quantum Nano-joint Thomson Elements Watanabe, T. (World Technology Instrument Co., Ltd.)	351
18. Investigation of the Effects of High-Deformation, High-Precision Manufacturing Processes on Material Properties Wilson, M. J.; Blobaum, K. J. M.; El-Dasher, B.; Florando, J. N.; Gross, R. J.; Seugling, R. M.; Swift, D. W. (Lawrence Livermore National Laboratory)	355
19. Micromixer of Multi-Material Slurry with Rotation-Split-Combination Cycle for Ceramic Components Yamauchi, T.; Nagato, K.; Ikeshima, S.; Hamaguchi, T.; Nakao, M. (The University of Tokyo).	359

Progress in Precision

1. Visual Inspection of Free Form Glossy Surfaces Using Phase Shifted Deflectometry Cilip, C. M.; Pratt, D. S.; Bauza, M. B. (Carl Zeiss Industrial Metrology LLC) and Packer, J. (Carl Zeiss Optical Inline Metrology)	363
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Poster Session II

Wednesday, October 24, 2012, 3:30 PM - 5:00PM

Multi-Axis Machine Design

1. Reconfiguration Method for Optimization of Volumetric Positioning Error of 3D Workspace Bui, C. B.; Nguyen, C. N. (Korea University of Science Technology) and Hwang, J.; Park, C.-H. (Korea Institute of Machinery & Materials (KIMM)).	367
2. Insertion of the Hohlraum into the TMP-Shell: Increasing Determinism, Yield, and Throughput Lawson, B. L.; Wang, I. A.; Myers, M. D.; Heinbockel, C. L.; Castro, C. (Lawrence Livermore National Laboratory)	371
3. Development Result of Double Arch-Type 5 Axis Machine Tool Lee, S. C.; Ro, S.-K.; Park, J.-K. (Korea Institute of Machinery & Materials (KIMM)).	372

Precision Component Design

1. Performance Analysis of Active Aerostatic Thrust Bearings with Dynamic Error Budgeting Aguirre, G.; Gorostiaga, M.; Buruaga, M.; Urreta, H. (IK4-IDEKO)	376
2. Design of a Deterministic Fixture for Handling CFTA's Used in Inertial Confinement Fusion Targets Buice, E. S.; Bennett, D. W.; Montesanti, R. C.; Reynolds, J. L. (Lawrence Livermore National Laboratory)	380
3. Precision Measurement of Nut Stiffness in Ballscrews Chung S.-C. (Hanyang University) and Kim, G.; Park, C.-H. (Korea Institute of Machinery & Materials (KIMM))	382
4. Mechanical Vacuum Feedthrough Using Gas Lubricated Guiding Elements Heidler, N. (Friedrich-Schiller University) and Gossel, M.; Risse, S.; Eberhardt, R. (Fraunhofer Institute for Applied Optics and Precision Engineering IOF) Schenk, C. (Vistec Electron Beam GmbH).	386
5. Modeling of Thermal Nut Stiffness in Ballscrews Min, B.-K.; Kim, M.; Chung, S.-C. (Hanyang University) Park, C.-H. (Korea Institute of Machinery & Materials (KIMM)).	390

6. Modeling and Uncertainty Estimation of Ballscrew Nut Stiffness	
Min, B.-K.; Cho, H.-G.; Chung, S.-C. (Hanyang University) and Park, C.-H. (Korea Institute of Machinery & Materials (KIMM))	394
7. Low Cost Hydrostatic Bearing Manufacturing	
Slocum, A. H.; Heberley, B.; Wong, A. R. (Massachusetts Institute of Technology)	398
8. Solar Power Concentrators: Bending Metal Tubing at High Precision	
Slocum, A. H.; Campbell, R. B.; Benjaminson, E. C. (Massachusetts Institute of Technology).	402
9. Element Level Optimization of a Cylindrical Flexure for Linear Motion Guidance	
Telleria, M. J.; Culpepper, M. L. (Massachusetts Institute of Technology)	406

Precision Machining Processes

1. Investigation of Ultra-Precision Turning Using Aerostatic and Hydrostatic Spindles	
Ahmar, E.; Brecher, C.; Wenzel, C. (Fraunhofer Institute for Production Technology) and Arneson, D. A. (Professional Instruments Company, Inc.)	410
2. Finishing of Electroless Nickel Plated Dies for Hard X-Ray Mirror Replication	
Beaucamp, A. T.; Matsumoto, A.; Namba, Y. (Chubu University)	414
3. A Study of Theoretical Modelling of Surface Generation in Computer Controlled Ultra-Precision Polishing of 3D-Structured Surfaces	
Cheung, C.-F.; Kong, L.-B.; Ho, Lai-Ting (The Hong Kong Polytechnic University)	418
4. Investigation of the Dynamic Grinding Forces Resulting from the Machine Compliance Leading to Regenerative Chatter	
Deakyne, T. R.; Marsh, E. R.; Keller, E. E. (The Pennsylvania State University) and Vallance, R. R.; Koh, K. H. (nanoPrecision Products, Inc.) and Knapp, B. R. (Professional Instruments Company, Inc.)	422
5. Ultra Precision Cutting of Tungsten Carbide Mold with PCD Milling Tool Fabricated by EDM and Grinding	
Furuki, T.; Suzuki, H. Y.; Okada, M. (Chubu University) and Fujii, K. (NS Tool Company, Ltd.)	426
6. Developing Capabilities for Fabricating Extended Polynomial Freeform Mirrors	
Jared, B. H.; Saavedra, M. P.; Sweatt, W. C. Boye, R. (Sandia National Laboratories)	430
7. Study on Machining High Aspect Ratio Micro Channel Structures via a Shaping Method Using a Diamond Tool	
Je, T.-J. ; Choi, D.-S.; Jeon, E.-C. (Korea Institute of Machinery & Materials (KIMM)) and Park, E.-S.; Choi, H.-J. (University of Science & Technology).	433
8. Real-Time Position Error Compensation for Slender Workpiece for Laser Machining	
Kim, B.-S.; Park, J.-K. (Korea Institute of Machinery & Materials (KIMM)) and Kim, Y.-P. (Hwacheon Machine Tool Co., Ltd.)	437
9. Effects of the Tool Edge Shape on the Brittle Cracking in the Glass Machining	
Ono, T. (Kyusyu Sangyo University)	441
10. FEM Analysis of the Effect of Sequential Cuts on Heat Partition in High-Speed Cutting	
Sasahara, H.; Shimomura, M. (Tokyo University of Agriculture and Technology).	445
11. Precision Grinding of Large Aspheric Glass Lenses for Digital Single Lens Reflex Cameras	
Suzuki, H. Y.; Kagohashi, Y.; Furuki, T.; Okada, M. (Chubu University)	449
12. Uniform Polishing of Large Aspheric Glass Lenses by Magnetic Field-Assisted Polishing	
Suzuki, H. Y.; Kato, D.; Okada, M. (Chubu University).	453

Precision Metrology

1. Simulation Studies of Sub-Micrometer Contact of Topography Styli Althagafy, K. T.; Chetwynd, D. G. (University of Warwick)	457
2. Simple Calibration Artifact for Use on Roundness Testers Arneson, D. A.; Liebers, M. J. (Professional Instruments Company, Inc.)	461
3. Design and Construction of a Precision Rotating Magnetometer Chao, L.; Afoakwa, R.; Zirkle, T.; Seifert, F.; Schlamminger, S.; Pratt, J. R.; Brandenburg, N (National Institute of Standards & Technology)	464
4. AFM Linewidth Measurement Employing 3-D Image Registration on Multi-View Scans Chen, L.-C.; Thai, N. V. (National Taipei University of Technology) and Liou, H.-C.; He, B.-C. (Industrial Technology Research Institute)	468
5. Reducing Uncertainty of Wettability Measurements Through Controlled Sessile Droplet Deposition: Current Issues Chesna, J. W.; Rivers, R. I.; Smith, S. T.; Her, T.-H. (University of North Carolina-Charlotte) and Samara, A. M. (Intel Corporation)	472
6. Overview of Axis of Rotation Centering and First Harmonic Errors for Fixed and Rotating Sensitive Direction Applications Dalrymple, T. M. (Dalrymple, Inc. and University of Florida)	476
7. CMM Measurement and Evaluation of Models with Trimmed Free-Form Surfaces ElKott, D.; Knebel, R. (Carl Zeiss Industrial Metrology, LLC) and Veldhuis, S. C. (McMaster University)	480
8. A Virtual Machine Tool for the Evaluation of Standardized 5-Axis Performance Tests Fesperman Jr., R. R.; Moylan, S. P.; Donmez, M. A. (National Institute of Standards and Technology)	484
9. Single and Multiple-Instrument Surface Characterization with the Area Structure Function He, L.; Evans, C. J.; Davies, A. D. (University of North Carolina at Charlotte)	488
10. Estimation of Volumetric Positioning Errors Using Three-Face Step-Diagonal Method Hwang, J.; Lee, C.-H.; (Korea Institute of Machinery & Materials (KIMM)) and Bui, C. B. (Korea University of Science and Technology)	492
11. Measurement and Characterization of 3D Aspherical Microlens Array for Advanced Optics Kong, L.-B.; Cheung, C.-F.; To, S. (The Hong Kong Polytechnic University)	496
12. Freeform Surface Analysis through use of Form Talysurf Lineberger, J. L.; Davies, M. A. (University of North Carolina at Charlotte)	500
13. Uncertainty of the Interferometric Radius Measurement in a Production Environment Medicus, K. M. (Optimax Systems, Inc.)	504
14. Evaluation Method of Acceleration Correlated Position Errors in Machine Tools Nguyen, M. H.; Weikert, S. (inspire AG) and Wegener, K. (Institute of Machine Tools and Manufacturing)	506
15. Dimensional Metrology of Plasmonic Nanolithography Machine Ni, K.; Hocken, R. J. (University of North Carolina at Charlotte)	510
16. Non-Contact Probe for On-Machine Metrology Parks, R. E. (Optical Perspectives Group, LLC) and Smythe, R. A. (R.A.Smythe, LLC) and Roblee J. W.; Jacobs, B. (AMETEK - Precitech, Inc.) and Stover, E. (AMETEK, Taylor Hobson Ltd.)	514

17. Experimental Characterization of a Flexible Capacitive Pressure Transducer
Petrzelka, J. (Massachusetts Institute of Technology) and Al-Qhatani, H. (King Fahd University of Petroleum and Minerals). 518

18 Performance Evaluation of Data Fusion Metrics
Ramasamy, S. K.; Raja, J. (University of North Carolina at Charlotte). 522

19. Density Measurement of Tantala (Ta2O5) Foams Using X-Ray Radiography
Seugling, R. M.; Young, P.; Sain, J.; Satcher, J. H. (Lawrence Livermore National Laboratory) 526

20. Geometric Error Simulation and Evaluation for Machine Tools
Shim, J.; Hwang, J.; Lee, C.-H. (Korea Institute of Machinery & Materials (KIMM)). 527

21. The Effect of Tip Size in Calibration of Surface Roughness Specimens with Rectangular Profiles
Song, J.-F. J.; Renegar, T. B.; Soons, J. A.; Muralikrishnan, B.; Villarrubia, J. S; Zheng, A. X.; Vorburger, T. V. (National Institute of Standards & Technology). 530

22. A New Successive Three Point Method of Roundness Measurements
Uda, Y.; Shimada, S. (Osaka Electro-Communication University) and Kiyono, S. (Precision Measurement Laboratory) 534



Technical Papers

CALIBRATION OF AREAL SURFACE TOPOGRAPHY MEASURING INSTRUMENTS: ARE WE THERE YET?

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INTRODUCTION

The answer to the question in the title is: no. But there has been significant progress recently and the end is in sight. This paper will discuss the progress made and address the steps that need to be taken before the answer is more optimistic.

TRACEABILITY AND CALIBRATION

An important issue that must be addressed to bring surface topography into compliance with manufacturing quality systems is the measurement traceability of the instruments. Whilst there is a traceability infrastructure for stylus instruments operating in a profile mode [1], there is not yet a satisfactory infrastructure for areal surface topography measuring instruments. To fill the traceability chain from the definition of the metre to an areal measurement in industry, a number of steps are required (see figure 1). Firstly, primary instrumentation that can measure areal surface topography is needed. A number of national measurement institutes (including NPL) have developed stylus instruments with displacement measuring laser interferometers that can determine the position of the stylus tip [2, 3]. Knowledge of the laser source wavelength in the interferometers assures traceability to the definition of the metre. Secondly, transfer artefacts are needed that can be calibrated via the primary instrumentation (either directly or indirectly) and be used in turn to calibrate instruments in industry.

NPL has been working towards a traceability infrastructure for areal measurement for a decade. A stylus-based primary instrument has been developed and methods for calculating uncertainties using this instrument have been devised based on a Monte-Carlo approach [4]. And NPL (with the University of Huddersfield) recently produced reference software for calculating areal surface texture parameters (at least for the areal field parameters [5] – work on the areal feature parameters is ongoing). NPL (with others in ISO/TC 213/WG 16) has also

developed a series of metrological characteristics which determine the scale calibration of an instrument [6]. The metrological characteristics, which should be common to all instrument types, are the noise and residual flatness [7]; the scale, linearity and squareness of the instrument axes [8]; and the resolution.

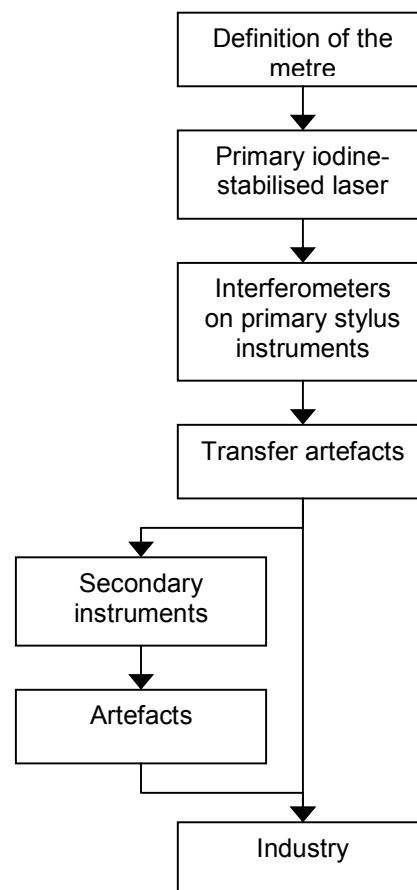


FIGURE 1. Flow diagram of the steps required for areal traceability to the metre. Note that in practice there would be another stabilised laser between the primary laser and the interferometers.

A series of transfer artefacts (known as the “Areal Bento Box”) have been developed to allow a user to determine the metrological characteristics of an instrument. Complex machining methods have been employed (for example, fast tool servo diamond turning at the University of Bremen and electron-beam lithography at the Karlsruhe Institute of Technology) to manufacture master artefacts, and nickel electroforming has been used to produce replicas (at Rubert + Co.). The artefacts include an optical flat, step heights, a series of crossed square wave gratings with different pitches, star patterns, a pseudo-random surface and a ball on a plane (see figure 2). The replication process allows the commercial cost of the artefacts to be kept at a minimum. NPL will now supply a set of calibrated artefacts along with a good practice guide that will allow the user to calibrate the metrological characteristics of a surface topography instrument, giving confidence in measurements taken with the instrument, and measurement uncertainties to be addressed in a limited number of measurement scenarios. To date, artefacts and good practice guides have been developed for stylus instruments; phase stepping and coherence scanning interferometers (CSI); and scanning confocal microscopes. Artefacts and guides for focus variation microscopy will follow shortly. However, the general philosophy behind the calibration scheme developed at NPL should extend to most of the instrument types that are commercially available.

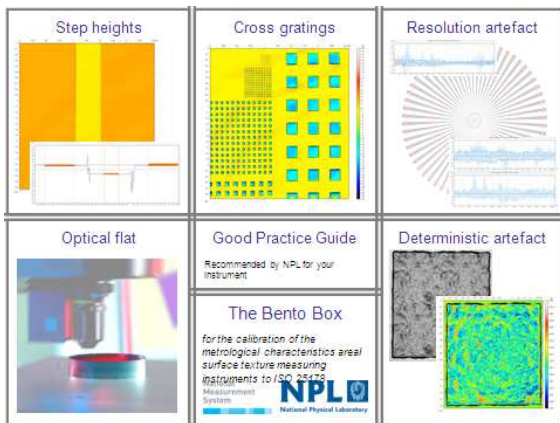


FIGURE 2. The NPL artefacts for calibration of areal surface topography measuring instruments.

In 2002, ISO technical committee 213 formed a working group (WG 16) to address standardization of areal surface texture measurement methods. WG 16 is developing a number of standards encompassing definitions of terms and parameters, calibration methods, file formats and characteristics of instruments. Several of these standards have been published and a number are at various stages in the review and approval process. At the time of writing there is much debate about how the different instrument types will be covered in the areal standards. On the one hand there is the opinion that one common specification standard is required that covers all instrument types (this opinion is shared by the authors), but on the other hand there is opinion that a separate standard is needed for each instrument type (this is the current situation). This debate must be settled soon.

A TRANSFER FUNCTION APPROACH

With the new ISO calibration scheme it is possible to determine the characteristics of the instrument scales. But, are real surfaces made up of a series of flat planes at different heights and spacings? Whilst the determination of the instrument scales is a step in the right direction, it does not calibrate the instrument to be used to measure a complex surface – for this the ability of the instrument to measure slopes on the surface must be characterized.

An instrument will have a finite spatial frequency response, i.e. it will transmit some spatial frequency components of the surface, it will block some components and others will be partially measured. If it is assumed that this process is linear, then the instrument simply acts as a linear filter with a specific transmission characteristic. To calibrate how the instrument responds to surfaces, the transmission characteristic needs to be determined. In the case of optical instruments, the transmission characteristic is given by the optical transfer function (OTF) which, in most of the cases encountered for optical surface topography measuring instruments, is given by the inverse Fourier transform of the point spread function (PSF). In turn, the PSF is the impulse response of the instrument and will contain information about the resolution of the instrument and any aberrations in the optics.

Calibration using the OTF gives information about how the instrument responds to slopes on

the surface. Although the PSF is by definition the response of the system to a point object, it cannot be measured as such. A point like object (or at least one that is significantly smaller than the PSF) is not easy to produce in practice and in any case it will violate the validity of the Kirchhoff approximation [9]. NPL and Loughborough University have developed a model for CSI based on weak scattering (known as the “foil model” [10]). In this case, a surface whose foil representation uniformly covers the spatial frequency domain is used as a calibration artefact, i.e. a sphere that is smaller than the field of view, but relatively large compared to the lateral resolution. It is worth noting that the sphere technique requires the form of the sphere to be calibrated with the same accuracy as is required of the instrument to be calibrated. This means that a method for measuring the form of spheres with diameters less than 100 μm is needed with nanometre accuracy. Current methods for small sphere form measurement (using larger reference spheres in the micro-CMM area) are only accurate to 40 nm [11], so a new method of sphere calibration is required. NPL is currently undertaking a project to develop a reference small sphere form measuring instrument. The foil model approach is now being applied to confocal microscopy and focus variation microscopy, and will be reported to ISO for potential standardization in due course.

In the case of confocal microscopy, the use of the OTF is not as straightforward as for CSI. The OTF contains information about modulation and phase. The PSF for CSI is an amplitude PSF because the source is incoherent, and therefore the phase information is preserved. For a laser scanning confocal microscope, the source is coherent, so the PSF is an intensity PSF, and the phase information is scrambled (due to the squaring of the amplitude). This means that a method needs to be developed to extract the phase from the intensity data (the confocal planes). NPL is currently working on this difficult theoretical problem.

Once a calibration method has been determined for a given instrument, it is then possible to correct (adjust) the instrument using the calibration data. NPL and Loughborough University have recently made corrections to a CSI instrument and the preliminary results can be seen in figure 3. Note that it was necessary to adjust the instrument for xy scale linearity and for optical distortion prior to the correction for the

OTF. A significant improvement in the measurement of a sphere is shown in the figure.

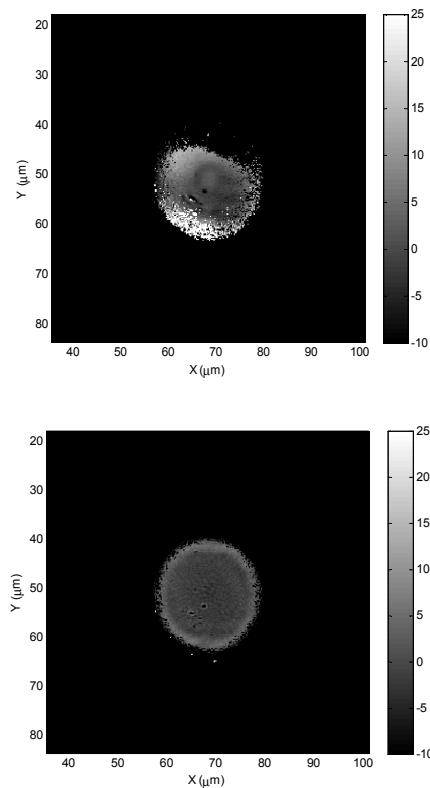


FIGURE 3. Measurement of a sphere using CSI. Top: before correction. Bottom: after correction.

The weak scattering assumption implies that there are no multiple reflections from the surface being measured – in most cases this in turn implies that there are no slopes on the surface that are outside the slope limitation of the numerical aperture. With rough surfaces, multiple reflections can be a significant source of error [12]. Also, if multiple reflections occur, the linear assumptions implicit in the transfer function models no longer apply. Further research is required firstly to calculate the magnitude of the measurement uncertainties caused by the various assumptions, and secondly to produce calibration techniques that apply to rough (strongly scattering) surfaces.

One aspect of calibration that has led to a great deal of debate over the last few years is resolution. The lateral resolution of an imaging system is not under debate – this is simply given by the Rayleigh limit of the Sparrow criterion, and is dependent upon the source wavelength

and numerical aperture [13]. There may be some adjustments to the values given for lateral resolution caused by effects such as that due to the pinhole in confocal microscopy, but such things are well-understood concepts in optics and highly predictable. However, a surface topography measuring instrument measures heights. Again, the axial (height) resolution is highly predictable (and usually much less than other axial uncertainty contributions [8]), but the ability of the instrument to measure height on a particular spatial frequency needs to be quantified [14]. To illustrate this, consider a square wave grating. The instrument may easily resolve the pitch of the grating, but as the grating wavelength approaches the value for the instrument's lateral resolution, the height information may not be faithfully measured. Each instrument will have a height transmission characteristic given by its transfer function. From the transfer function it is then possible to determine an optical cut-off frequency at a given height modulation (e.g. ISO is considering 50 %).

NPL is currently developing methods for the determination of optical cut-off using a range of techniques including the use of small spheres, step heights and star patterns. This work will be presented in later publications.

DISCUSSION

A recent comparison of a stylus instrument and a CSI has shown that it is possible to get the two instruments to agree on a range of surfaces [15]. However, care has to be taken to ensure that the spatial bandwidths that the instruments sample are comparable [16]. Using the techniques discussed in this paper to determine the ISO metrological characteristics will allow some comparability (for length measurements at least) of instruments, but further research and standardization steps are required. Firstly, the transfer function of the instrument needs to be determined. This will allow calibration and instrument adjustment in the weak scattering case (i.e. when there is no multiple scatter). Secondly, methods for calculating the measurement uncertainty associated with the transfer function approach need to be developed, clearly taking into account any mathematical assumptions required by the models.

Instruments for measuring areal surface topography have been around for several decades. But it is only recently that methods for calibration of the instruments have been developed that are not based on gross simplifications (e.g. the use of a single step height artefact to calibrate an instrument that will be used to measure complex surfaces). It is generally accepted that many of the optical instruments have nanometre axial resolution. With the methods for instrument calibration and adjustment discussed in this paper, it is now possible to demonstrate nanometre accuracy.

ACKNOWLEDGEMENTS

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Computed Tomography in Metrology

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INTRODUCTION

Computed Tomography (CT) was introduced in 1972 by Hounsfield and Cormack as an innovative departure from traditional X-ray technology used in medical applications and efforts in that field were awarded a Noble Prize in medicine [1]. The origin of the name comes from Greek tomos (slice) and graphein (to write). A CT image provides a three – dimensional representation of an object derived from a series of two – dimensional images combined into a 3D representation of a part or assembly, FIGURES 16, 17, 19. While the first CT image took nine days to be produced, today it may take only seconds to collect the data and reconstruct a 3D image. With the rapid increase in computing power, it will only get faster. Over the past 10 years CT started migrating from medical applications to industrial metrology applications. However, metrological instruments require quite a different design and data analysis approach to produce measurements that are repeatable and provide a useful uncertainty budget as well.

PRINCIPLE OF OPERATION

X-rays are a form of electromagnetic radiation with the wavelength range from 0.01 to 10 nanometers and frequency above 30×10^{15} Hz. Furthermore, X-rays are produced when high velocity electrons emitted by a hot cathode collide with a metal target - an anode. Finally, the energy emitted by an X-ray photon is produced from the energy of the colliding electron. However, it never exceeds the energy of the electron itself. A typical X-ray source produces a band of X-ray frequencies originating from the area on the target called the spot. The spot size is dependent on the energy and power output of the very source: the higher the power output, the larger the spot, FIGURES 3 and 4. A typical spot for μ -CT source is 3-5 μ m in diameter at a lower power output.

The projection CT system adopted for metrological applications consists of an X-ray source, a rotary axis, motion stages, and an X-ray detector, FIGURE 1. The image acquisition

is based on the ability of X-rays to penetrate objects and the amount of absorbed radiation is dependent on the shape and type of material.

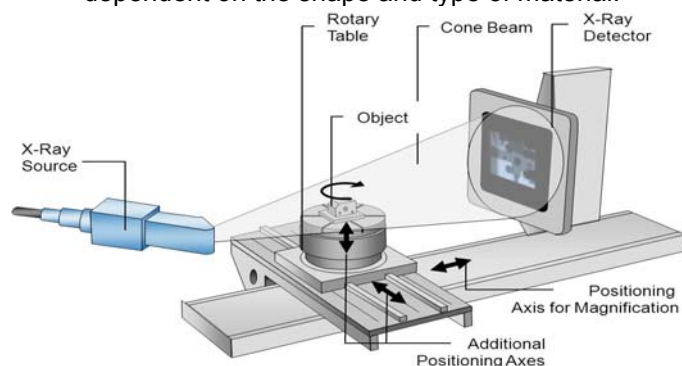


FIGURE 1. Schematic of projection CT system.

The X-rays, which penetrate through the object, are detected by a digital detector panel and a 2D image is formed. By placing a part on a rotary stage and repeating the process at each angular position, a series of 2D images are produced, FIGURE 2. Next, through computed reconstruction, a 3D image is obtained based on 2D views and the angular position of the object at the moment of image creation. As a result, the CT scan displays the part as a three-dimensional image composed of voxels [2].

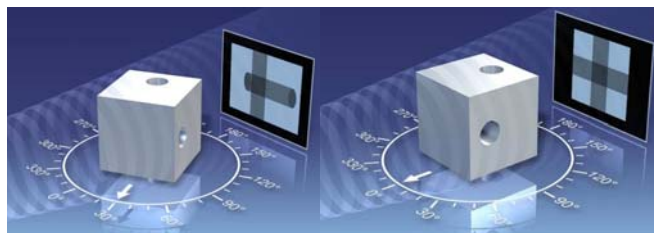


FIGURE 2. 3D image creation from series of 2D images.

Each voxel has a gray value that represents the local X-ray absorption density. The resolution of this 3D imaging system is determined in part by the number of voxels in the 3D image, which in turn depends on the number of pixels in the detector. By using sub-pixel respectively sub-voxel interpolation, the effective resolution of edge detection can be increased. However,

features smaller than the base pixel will not be visible. Using techniques derived from 2D image processing, 3D voxel images can be evaluated to achieve dimensional measurements for the scanned parts.

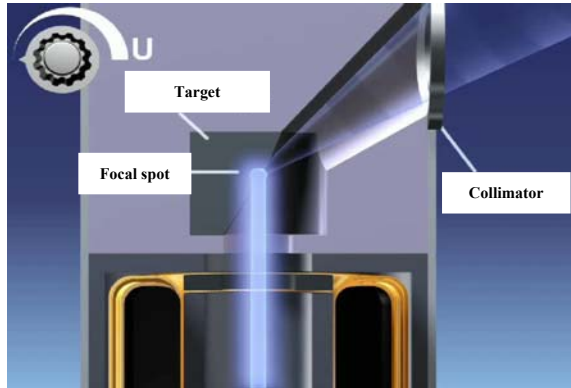


FIGURE 3. Source spot size at low power.

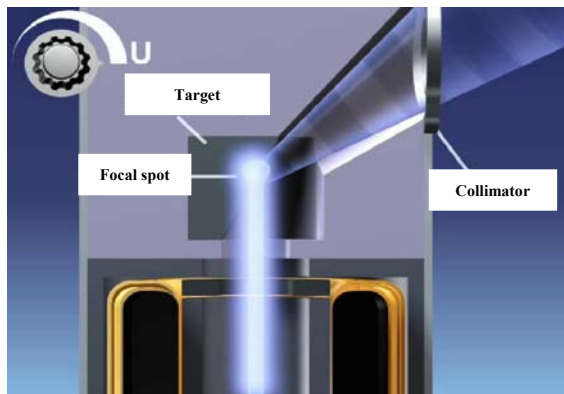


FIGURE 4. Source spot size at high power.

The combination of metrology and CT allows us to perform quick and nondestructive measurements on parts with difficult to reach structures. It is no longer necessary to cover the parts with synthetic resin and gradually destroy them to measure two-dimensionally plane by plane. In addition, a CT CMM performs measurements without exerting a measurement force that might deform soft materials. The complete 3D capture of a workpiece makes it possible to compare CAD models and measurement data sets, FIGURES 16 and 17. The color-coded result is an easy to interpret display of a geometry comparison, which provides meaningful information about the dimensional and form characteristics of the entire workpiece.

METROLOGICAL CT - REQUIREMENTS

Metrological CT, besides delivering X-ray images as a classic CT system would, has to generate those images with accurate scale and position with respect to each other. A typical high accuracy CT scan takes 30-90 minutes and during this time a series of 2D images are taken. It is critical that the alignment between source, measured part, and X-ray detector stays stable during the measurement, FIGURE 6. Furthermore, this stability is critical not only during the measurement but over an extended period of time to ensure the instrument's repeatability and reproducibility. Within this context it should be noted that classical CT systems provide great flexibility of part movement and alignment; however at great expense of the system's stability, repeatability, and overall accuracy.

The fidelity of the X-ray detector is critical as well. As mentioned above, the image is created by casting an X-ray "shadow" on the detector plane. Next, based on physical position of individual pixels, the image is created and dimensional relations between each data point are recorded. Unfortunately, the digital panels currently available are not uniformly produced to required fidelity and therefore a series of corrective actions are required. It is common that an X-ray detector panel has pixels placed with large uncertainty relative to their desired true location. Furthermore, the detector panel might be missing single pixels as well as entire rows, FIGURE 5. This is not a problem for qualitative analysis; however, it poses a big challenge for quantitative analysis. Moreover, each pixel might have a different sensitivity value causing significant image distortions. To compensate for all of these issues, each panel has to be carefully inspected and mapped for pixel position and sensitivity and adjustments have to be made for possible missing pixels.

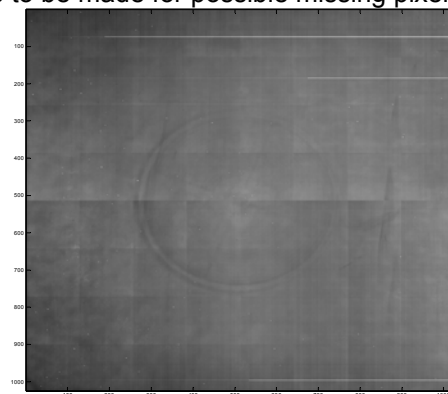


FIGURE 5. Typical detector defects.

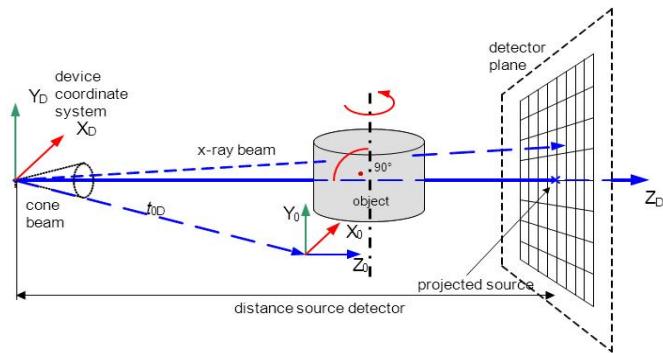


FIGURE 6. Geometrical requirements for metrological CT.

X-ray source spot size defines how wide a shadow of the scanned part's edge will be. However, the resolution of the image is also dependent on the pixel size of the detector, FIGURE 7. Due to the basic physics of shadow casting on a digital panel, ultimately the pixel size will limit the image resolution, make the spot size smaller and smaller, and not have a significant effect on the resolution ability of the CT machine, unless the pixel size is reduced as well.

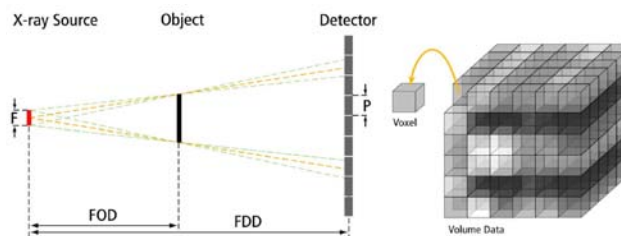


FIGURE 7. Voxel resolution dependent on spot size and detector pixel size. (Source: GE)

Given the fact that the appearance of the scanned component's edge is murky, the question is how to repeatedly detect the edge, which is not a concern in classical (qualitative) CT. The edge detection process starts by defining the "solid" and "air" components of the image. Then, the gray scale is defined by all the gray value between what is already described as "solid" and "air". For simplicity, we will assume that the edge is at 50% value between those two extremes, FIGURE 8. Based on the image, it is clear that 50% falls between two discrete steps of gray; therefore an interpolation technique can be applied to "enhance" the resolution. Unfortunately, this technique is not very accurate, since measured cross-sections are not uniform and various thicknesses will result in

different shades of gray representing a measured "solid", FIGURE 9.

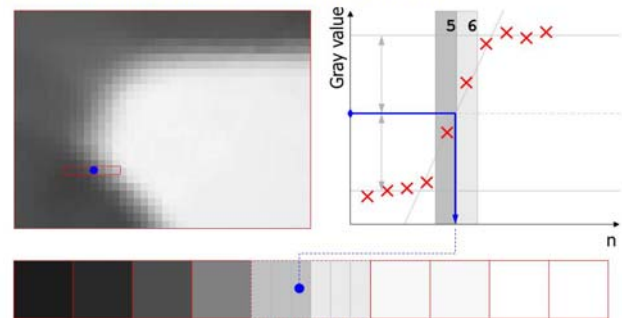


FIGURE 8. Edge detection and interpolation limits.

Given this phenomenon, it is necessary to localize the boundary definition and assess independently the boundary condition for the "solid" and "air" components of the image at each thickness / density variation. Using a nominal CAD model of the measured object can enhance this process and predetermine regions of various thicknesses. FIGURE 9 shows the effect of the global "non-corrected" approach for edge detection and adaptive (localized) edge definition. In this example the improvement is eightfold.

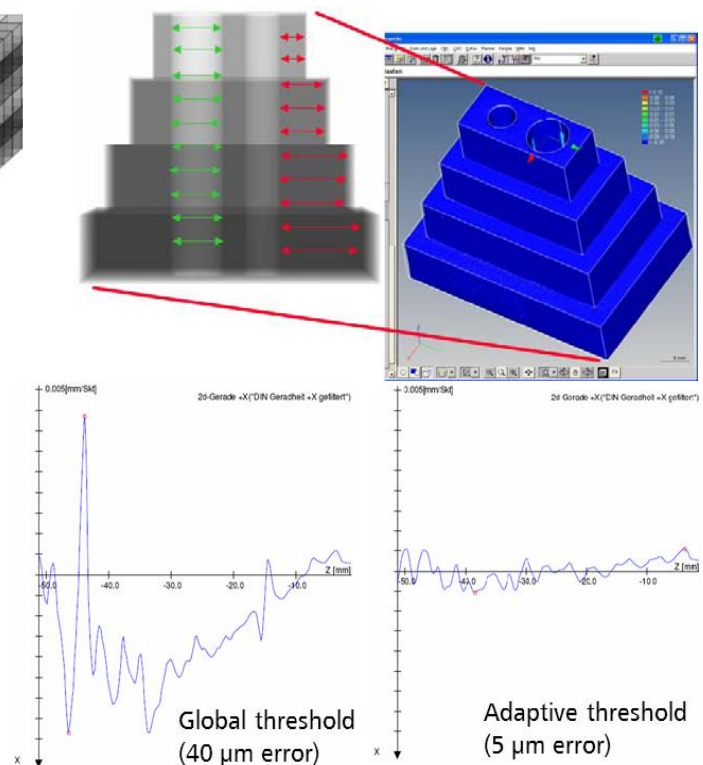


FIGURE 9. Effects of threshold method on edge detection accuracy.

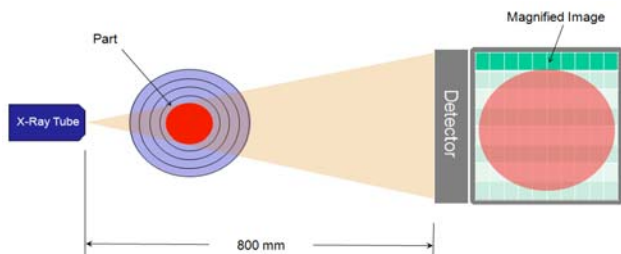


FIGURE 10. CT – variable resolution.

It is important to note that interpolation should only be applied to edge detection. The ability of the system to realize the smallest feature size will still be driven by the combination of the pixel size and projected image amplification. Therefore, the smallest feature size, which can be reliably detected, should be defined as three times bigger than the basic resolution derived from pixel size and projection amplification.

As mentioned above, the resolution is defined by three components: the spot size of the X-ray source, the pixel size on the X-ray detector, and finally the distance ratio between source and measured object, as well as the detector and measured object. Therefore, the closer the object is placed to the source, the greater the magnification. If the source produces a constant power output, which in turn results in a constant size of the spot, then the size of the measured object would define what resolution can possibly be achieved when the entire part is scanned at once. The bigger the object, the further from the source it has to be placed to capture its entire envelope and this results in lower resolution. It is possible to create a higher resolution image of a relatively large object by moving it closer and focusing on a region of interest. As a result, high resolution images can be stitched to the low resolution image revealing a great level of detail, FIGURE 14. However, the reader should keep in mind that if the object is placed too close to the source, the obtained image will be noisy and expected gain in resolution will be lost.

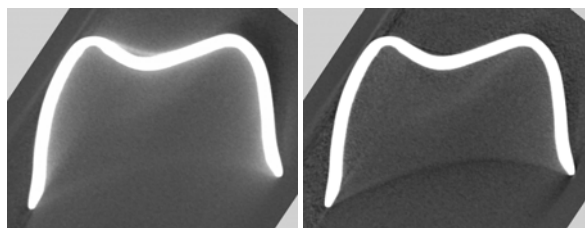


FIGURE 11. Cross-section of a dental implant (zirconium ceramic) with beam hardening effect (left) and corrected image (right).

Beam hardening is an example of a process-related effect which is traced to the multi-frequency nature of the X-ray spectrum used during image creation [3]. Various frequencies have different abilities to penetrate an object which is composed of multiple thicknesses and geometries. However, with advanced correction techniques, it is possible to successfully overcome this issue and produce an undistorted image of a measured component, FIGURES 11 and 16. It should be noted that correction for beam hardening has to be performed very carefully; otherwise it might result in a loss of contrast [4].

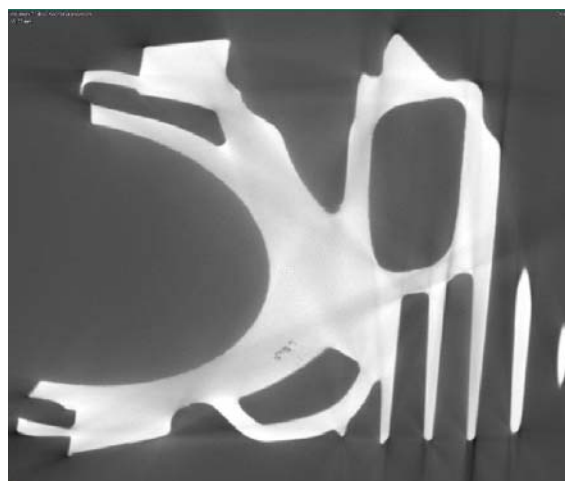


FIGURE 12. Cross-section of an aluminum part with beam scattering effect.



FIGURE 13. Cross-section of an aluminum part with corrected beam scattering effect.

Another phenomenon which greatly influences image quality is coherent scattering of an X-ray. This occurs when the X-rays interact with the object's atomic structure causing the excitement

of exposed atoms. As a consequence, those atoms become secondary sources of X-rays, FIGURES 12 and 13. Furthermore, such an X-ray source will have the same energy as the primary source and therefore the same wavelength. It is very important for the metrological system to be able to correct for such distortion in order to avoid a significant accuracy loss.

Reconstruction of a CT image is very computation intensive. Therefore it is common practice in qualitative analysis to reduce the number of data points through compression. Metrological applications are very sensitive to the fidelity of the collected data points, and through compression some of this fidelity will be compromised. Therefore, to produce accurate measurements the reconstruction should be performed on non-compressed data sets.

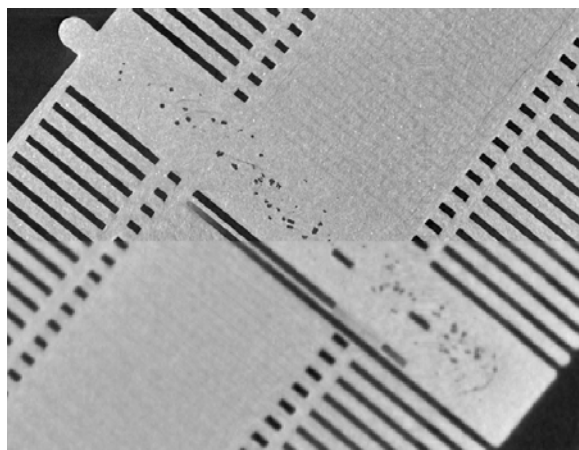


FIGURE 14. Effects of accurate high and low resolution data stitching based on scans with metrological CT.

The basic goal for any metrological CT system is to provide consistent, repeatable and accurate measurements which can be traced to a known artifact. The above described X-ray imaging characteristics are highly non-linear in nature. For these reasons, it is very difficult to “predict” the accurate location of internal features based on their relative expected position with respect to known external features previously measured with other probing systems.

Currently, metrological CT systems can perform successful measurements with sub micrometer interpolated resolution of edge detection. Those measurements can be performed with accuracy better than 2 μm . FIGURE 15, shows measurement deviation from a calibrated value

of a distance between selected spheres. The artifact was calibrated using a scanning CMM and the uncertainty of this calibration was 1 μm .

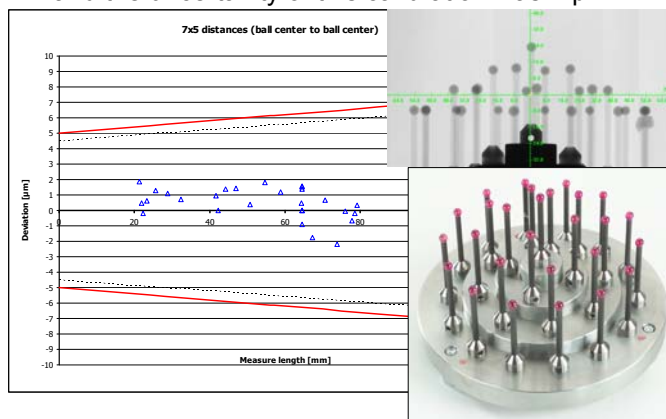


FIGURE 15. Metrological CT – Metrotom 800 the accuracy of distance measurement between multiple spheres based on VDI/VDE 2630 standard.

APPLICATIONS

Today Computed Tomography has revolutionized the industry by providing unique information regarding the metrological aspects of complex structures and inaccessible features. Applications span from consumer products to electronics, from the automotive and aircraft to the medical devices industry. With the aid of modern CT, the reduction of R&D time is significant, resulting in faster first part inspection or shorter reaction times on failure analyses. As a result, CT plays an important role in many critical decisions in daily industrial manufacturing and process control. For example, the plastics industry has changed their testing and manufacturing processes based on the capabilities of this promising technology.

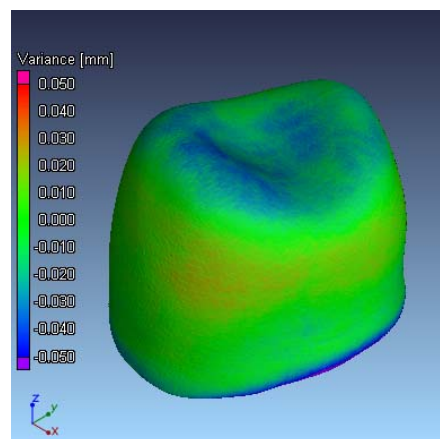


FIGURE 16. 3D representation of measured dental implant shown in FIGURE 11.

However, confidence in measurement results produced by CT systems is still low (due to qualitative CT systems) and a methodological approach that reflects a comprehensive understanding of the boundaries between qualitative and quantitative computed tomography is needed.

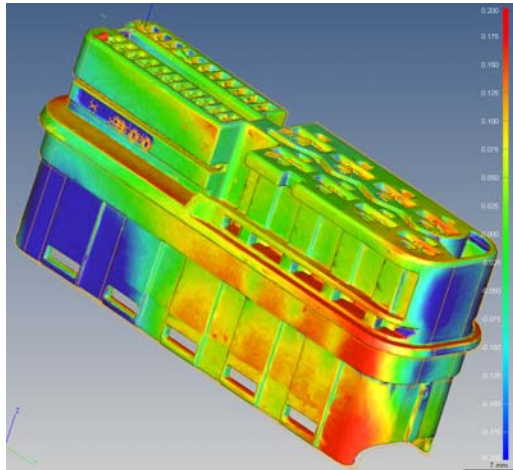


FIGURE 17. Color-coded CAD comparison of an electric connector block.

Complex parts with several thousand features can be measured and analyzed in hours instead of weeks. The obtained data can be compared with nominal dimensions and color coded deviation maps can be created, FIGURE 17.

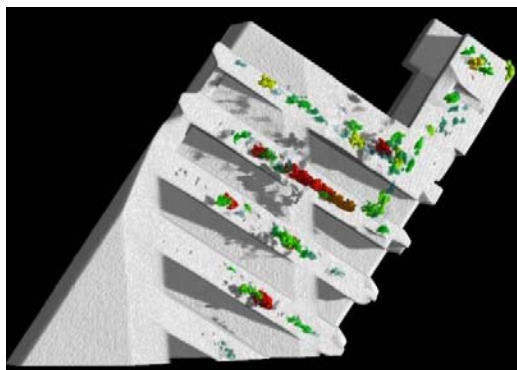


FIGURE 18. 3D porosity analysis.

Size and distribution of porosity in components often compromises the part's usability. With metrological CT size and location of each pore can be assessed and depends on the preset threshold value, obtained deviations can be color coded for better image clarity, FIGURE 18.

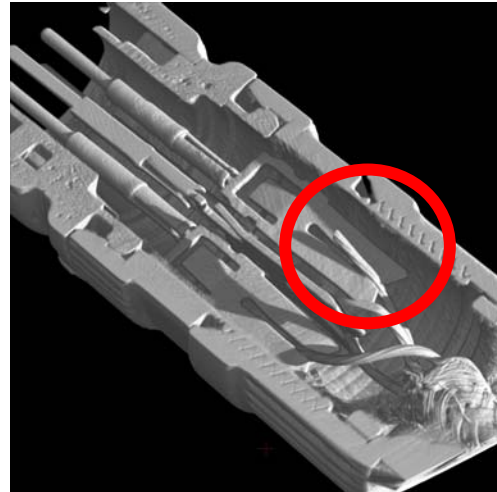


FIGURE 19. Assembly analysis of an electrical connector showing potential point of future failure.

Complex assemblies can be analyzed and defective processes can be adjusted based on qualitative data. Points of potential failure can be detected and operational failure of an assembled component prevented, FIGURE 19.

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SYNTHESIZING PRECISION FLEXURES THAT DECOUPLE DISPLACEMENT-BASED ACTUATORS

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INTRODUCTION

The purpose of this work is to generate a deterministic approach for synthesizing flexures that decouple the actuators of multi-degree of freedom (DOF) precision motion stages. This approach utilizes the geometric shapes of the FACT synthesis approach [1] to help designers rapidly visualize and consider every flexure topology that may be used to decouple any set of displacement-based actuators (e.g., set-screw, piezo, thermal, and magnetostrictive actuators). The ability to correctly synthesize such flexures is important because they (i) improve the controllability of the system's stage by not allowing the output of one actuator to affect the output of the other actuators, (ii) minimize the stage's parasitic motion errors by only transmitting the intended motions through the flexures to the stage over the full stroke, and (iii) increase the life of each displacement-based actuator by reducing undesired loads (e.g., shearing, bending, or twisting loads) imposed by one actuator to the other actuators. This work most directly impacts the synthesis of large-stroke, low-speed, multi-DOF precision flexure systems that are driven by displacement-based actuators.

MOTIVATION

The problem of effectively synthesizing flexures that decouple displacement-based actuators has been a challenge for precision motion stage designers for decades. Although some designers have successfully generated concepts that optimally satisfy their design's specific functional requirements [2], no general systematic approach currently exists for helping designers consider every concept, which would optimally decouple the displacement-based actuators within any multi-DOF system.

FUNDAMENTAL PRINCIPLES

The principles of the FACT synthesis approach have been adapted to help designers consider every way any set of displacement-based actuators may be decoupled such that they may

displace the system's stage large amounts without imposing damaging forces on other actuators. Before discussing how FACT may be applied to the synthesis of decoupled systems, consider first the three DOF system shown in Fig. 1A. The stage of this system is constrained by four bent blade flexures that permit the stage to move with two in-plane translations and one in-plane rotation. These DOFs are actuated by three piezo actuators shown as blue blocks in the figure. Every combination of these three DOFs may be achieved by simultaneously actuating these three actuators with various magnitudes. The actuators of the design from Fig. 1A are not decoupled. As one actuator expands, damaging transverse forces are imposed on the other actuators. These damaging forces become more problematic the farther the stage is displaced.

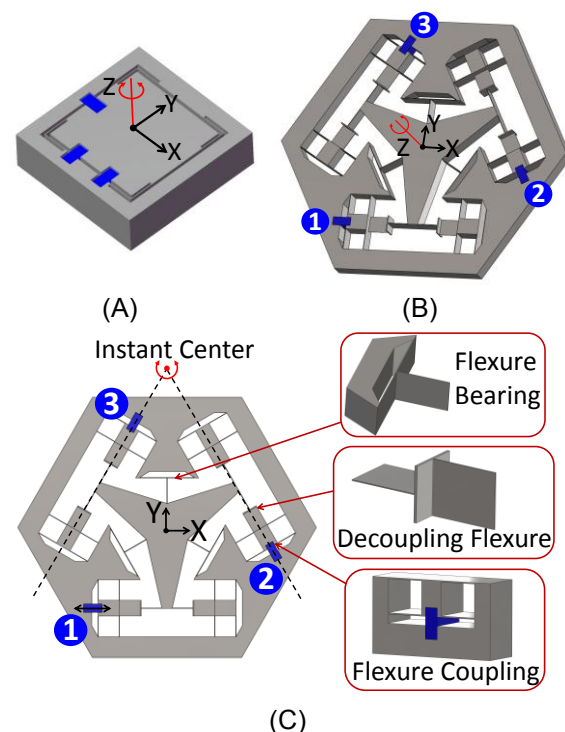


FIGURE 1. Three DOF flexure systems with coupled (A) and decoupled (B-C) actuators

Now consider the decoupled system shown in Fig. 1B-C, which was designed using FACT. This system would possess the same DOFs as the system in Fig. 1A, but its actuators would not experience the same undesired loads. If actuator 1 were actuated, the stage would rotate about the instant center shown in the figure independent of the two other rotations caused by actuators 2 and 3. Like most decoupled systems, the flexure topology of this system consists of three types of flexural elements as shown in Fig. 1C—flexure bearings, flexure couplings, and decoupling flexures. The job of flexure bearings is to guide the stage with its intended DOFs while constraining it to not move in other directions. The job of flexure couplings is to make sure that the stationary actuators are not subjected to any undesired loads except for those along their intended lines of actuation. The job of decoupling flexures is to prevent transmission of the motions induced by other actuators in the system to the flexure couplings of their corresponding actuator while permitting their corresponding actuator to stiffly transmit its intended motion to the system's stage.

Rules have also been created and integrated into the FACT synthesis approach for determining how much constraint each flexural element could contribute to the system for controlling exact, over, or underconstraint. Note that for the design of Fig. 1B-C, each serial actuation chain (i.e., the flexure chains consisting of the combined flexure couplings and decoupling flexures) is a six DOF flexure and thus provides no constraint to the stage. Thus, if the actuators were removed, the stage would be exactly constrained by the three flexure bearings to only possess the desired three DOFs. If each of the three actuators were locked in place such that their corresponding flexure couplings could not move, the stage would be totally exactly constrained and would not be able to move.

Not all decoupled flexure systems require separate flexure bearings to guide their stage with the desired DOFs like the design of Fig. 1B-C. FACT has also been adapted to enable designers to synthesize decoupled flexure systems that possess serial actuation chains, which not only decouple the system's actuators, but simultaneously act as bearings that guide the stage with the desired DOFs. Examples of such systems are shown in Figs. 2 and 3.

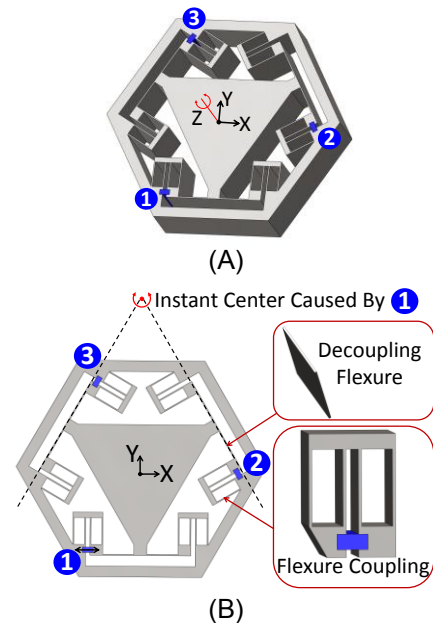


FIGURE 2. Another decoupled design (A) with its flexural elements labeled (B)

The biggest difference between these two designs is that the three actuators in the design from Fig. 2 are each uniquely linked to three instant center rotations, which may be combined to achieve the three desired DOFs (i.e., the two translations along the x- and y-axes and the rotation about the z-axis), whereas the three actuators in the design from Fig. 3 each directly link to these DOFs. If we wished the stage of the design from Fig. 2 to translate along the x-axis, actuators 1, 2, and 3 should be actuated with a magnitude ratio of 2:-1:-1 instead of 1:0:0 as would be the case for the design of Fig. 3.

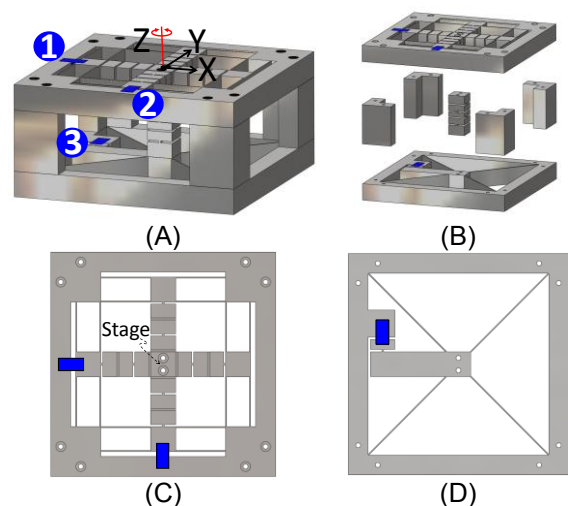


FIGURE 3. Another design (A) with exploded view (B), top piece (C), and bottom piece (D)

SYNTHESIS APPROACH

This section explains how FACT may be applied to the synthesis of decoupled flexure systems. To better understand the principles of FACT, however, we will first analyze the design of Fig. 2 and show how its serial actuation chains not only decouple its actuators, but also act as bearings to guide the stage with the desired DOFs. Figure 4 shows one of these serial actuation chains. The freedom space [1] of its flexure coupling is a single translation arrow along the axis of the actuator. The freedom space of its decoupling flexure is a plane of rotation lines and an orthogonal translation arrow. These freedom spaces are geometric shapes, which represent the DOFs that their flexural elements permit. To determine the total kinematic effect of this serial actuation chain on the stage, we must linearly combine these freedom spaces [1]. The resulting freedom space is shown in Fig. 4B and represents every rotation line on the plane of the flexure blade, every rotation line that is normal to the system's stage as shown by the box of parallel lines, and every translation arrow that points in directions, which are orthogonal to these lines as shown by the disk of arrows in the figure. According to FACT, this freedom space uniquely links to a constraint space, which consists of every parallel constraint line that lies on the plane of the flexure blade and is parallel to the rotation lines in the box of its freedom space as shown in Fig. 4B. If wire flexures were to be selected from this constraint space as shown in Fig. 4C, this new topology would represent the effective constraint that the serial actuation chain from Fig. 4A imposes on the stage (i.e., the topologies of Fig. 4A and Fig. 4C contribute equivalent kinematic/constraint characteristics to the system). Note that if this effective constraint topology of wire flexures replaced every serial actuation chain from the system of Fig. 2, the constraint space for the entire system would consist of every constraint line, which is normal to the surface of the stage as shown by the box of parallel lines in Fig. 4D. This constraint space uniquely links to a freedom space, which consists of all rotation lines that are parallel to the constraint lines as well as all translation arrows that are orthogonal to these lines as shown in Fig. 4E. This conclusion demonstrates that the flexure topology of the system from Fig. 2 does behave as a bearing to guide the stage with the freedom space of Fig. 4E, which contains the three desired DOFs as shown in the figure.

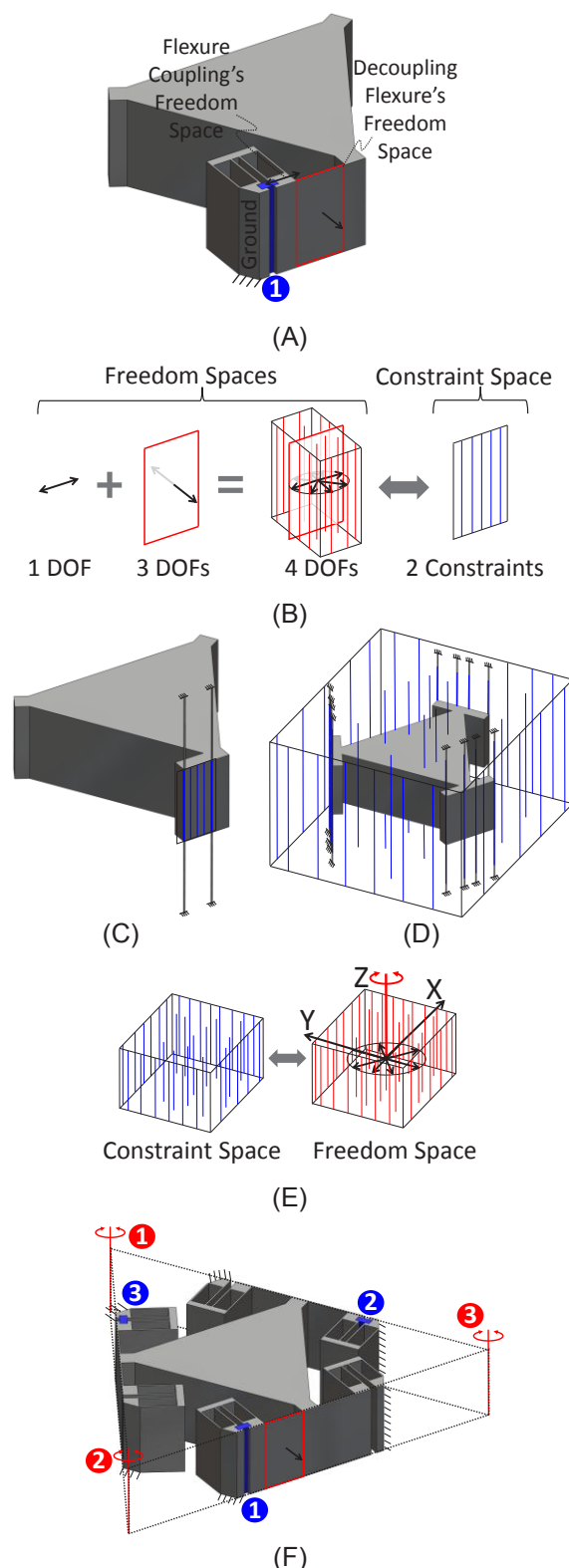


FIGURE 4. Serial actuation chain freedom spaces (A), effective constraint space (B), effective constraints (C), total system constraint space (D), total system freedom space (E), rotation lines lie within planar freedom space (F)

The shapes of FACT may also be applied to confirm that the serial actuation chains are properly decoupling the system's actuators. *If the freedom space of the chain's decoupling flexure contains the motions caused by the other actuators in the system, then the system's actuators will be properly decoupled.* Note that the two rotations caused by actuators 2 and 3 shown in Fig. 4F do lie in the plane of rotations (i.e., the freedom space) of the decoupling flexure pertaining to actuator 1. Note also from Fig. 4A that this freedom space does not contain the translation arrow of its flexure coupling. If this were the case, the actuator would not be able to stiffly transmit its intended motion to the stage.

We are now ready to apply the principles discussed previously to establish the systematic steps that may be used to consider every way any decoupled flexure system may be synthesized using the shapes of FACT.

Step 1: Identify the freedom space that contains the desired system DOFs.

Step 2: Identify the desired motions within that freedom space that are to be actuated and select an appropriate actuation scheme for actuating those motions.

Step 3: Specify the freedom spaces of those actuators and use their constraint spaces to synthesize appropriate flexure couplings.

Step 4: Select effective constraint spaces from within the constraint space of the freedom space selected in Step 1.

Step 5: Select a freedom space that lies within each effective constraint space's freedom space, which (i) does not contain the freedom space of its actuator, (ii) linearly combines with this freedom space to become the freedom space of its effective constraint space, and (iii) contains the motions of the other actuators.

Step 6: Synthesize the decoupling flexures using the constraint spaces of each freedom space from Step 5.

RESULTS

The system of Fig. 2 was fabricated (see Fig. 5A) and tested to measure how well its actuator outputs were decoupled by its flexure topology. An Opg Smart Scope Flash 200 Optical Comparator was used to measure the stage's parasitic errors in the y-axis as it was driven along the x-axis and the results are shown in Fig. 5B. Note that over a range of 1mm in the x-axis, the trend of the stage's error in the y-axis is nearly a horizontal line with a slope of

0.0012mm/mm. The small slope of this trend line demonstrates the success of the decoupling flexures.

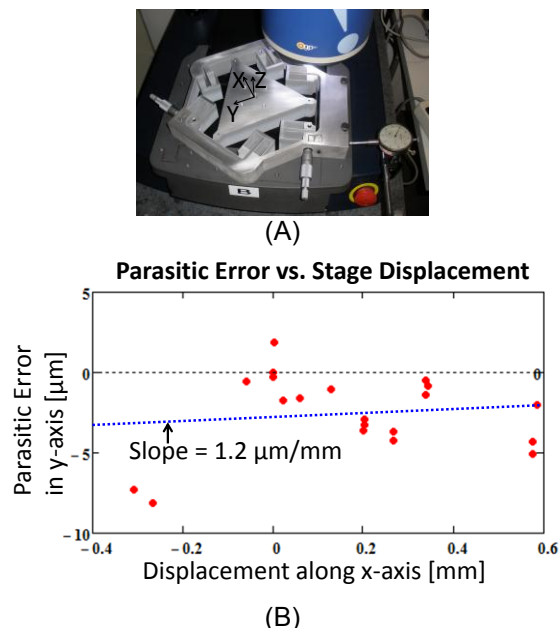


FIGURE 5. Fabricated flexure (A) and data showing coupled drift in an unwanted axis (B)

CONCLUSIONS

The principles of FACT have been adapted to help designers synthesize flexures that decouple displacement-based actuators such that they (i) do not impose undesired loads on other actuators, (ii) do not affect the outputs of other actuators, and (iii) drive the system's stage to move with desired DOFs and minimal parasitic error. Rules have been created for controlling system constraint characteristics (i.e., exact, over, and under-constraint). A FACT-designed flexure system was fabricated and tested to experimentally verify how well its flexures decoupled the outputs of its actuators. This work was performed under the auspices of the U.S. Department of Energy by Lawrence Livermore National Laboratory under Contract DE-AC52-07NA27344.

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WATER-COOLED HEAT LOAD CHOPPER FOR HIGH-ENERGY SYNCHROTRON EXPERIMENTS

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INTRODUCTION

The Advanced Photon Source (APS) at Argonne National Laboratory is one of five US national synchrotron radiation light sources. A beam of high-energy electrons at APS emits synchrotron radiation in wavelengths suitable for imaging features of atoms, molecules, crystals, and cells. The APS currently provides the brightest high-energy X-ray beams in the Western Hemisphere [1].



FIGURE 1. Aerial photo of the APS. The storage ring has a circumference of 1.1 km. Photo Credit: Argonne National Laboratory.

At BioCARS in the Center for Advanced Radiation Sources at APS is dedicated to time-resolved macromolecular crystallography. Traditional X-ray crystallographic techniques provide a static view of atomic structure. Time-resolved X-ray crystallography can provide insight into the mechanisms of molecular function.

In addition to the setup shown in Figure 2, time resolution at BioCARS is provided by precise synchronization of two rotating-disk choppers and a shutter [2].

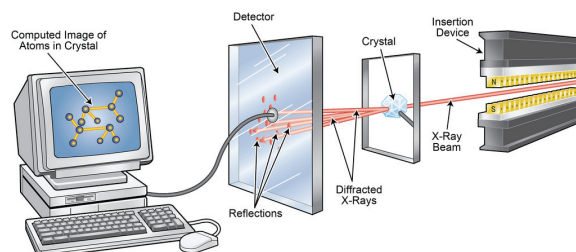


FIGURE 2. Traditional X-ray scattering experiment setup. At BioCARS, the choppers are located before the crystal. Photo Credit: Argonne National Laboratory.

The first chopper, a water-cooled heat load chopper, reduces the average power of the X-ray beam to protect sensitive equipment and samples from damage. The high-intensity bursts of pulses exit the heat load chopper and travel through a shutter which isolates a single burst of synchrotron pulses. Finally, a high speed chopper is used to extract a single or multiple 100 ps X-ray pulses from the isolated burst. The timing of each chopper is synchronized with the storage-ring reference clock at 351.93 MHz. This paper describes the design and testing the water-cooled air bearing heat load chopper recently installed at BioCARS Sector 14 [2].

CHOPPER DESIGN

Instead of attenuating the X-ray beam intensity to a level that safe for the high speed chopper and the samples, the average power is reduced by controlling the duty cycle of the beam. The heat load chopper shown in Figure 3 is designed to block a 1.5 mm, 520 W X-ray beam that has a power density 4.7 times greater than the surface emission of the sun [2].

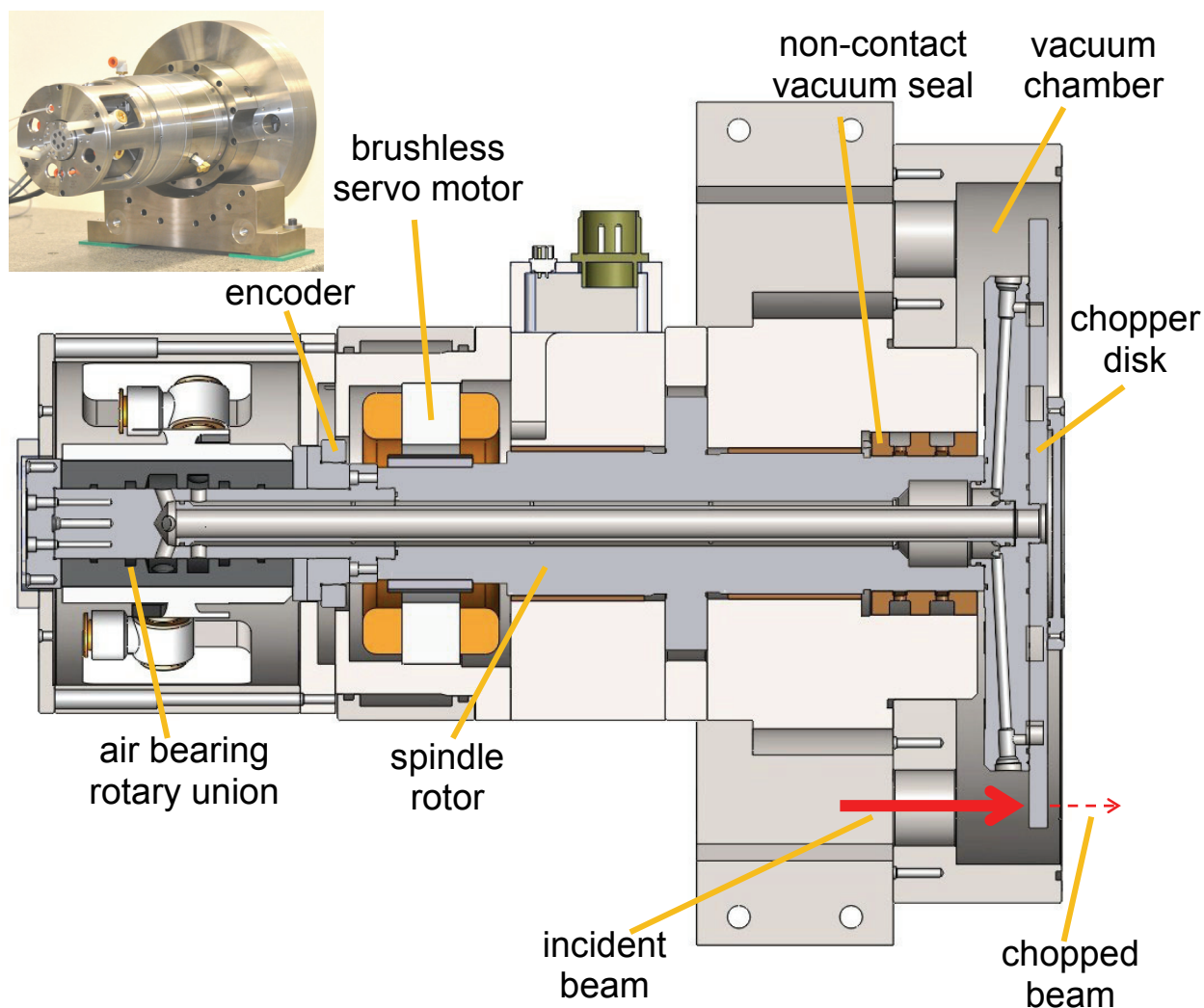


FIGURE 3. Cross-section of the water-cooled air bearing chopper spindle. The duty cycle of the high intensity X-ray beam is modulated by the rotating water-cooled chopper disk.

The beam is chopped by a slotted disk that spins in a vacuum at nominally 5000 RPM. Vacuum levels as low as 5×10^{-6} Torr have been achieved in the vacuum chamber via differentially pumped annular vacuum seals. These non-contact seals are designed to provide a jitter-free alternative to traditional seals.

The 275 mm diameter 416 stainless disc shown in Figure 4 is 7.6 mm thick and the calculated attenuation at 12 keV is 7×10^{-21} [2]. Various slot widths in the disk allow transmission between 0.2% and 1.6% duty cycle to protect downstream equipment and samples. At the minimum duty cycle of 0.2%, the chopper reduces the average power from 520 W to about 1 W. The reduced heat load on downstream

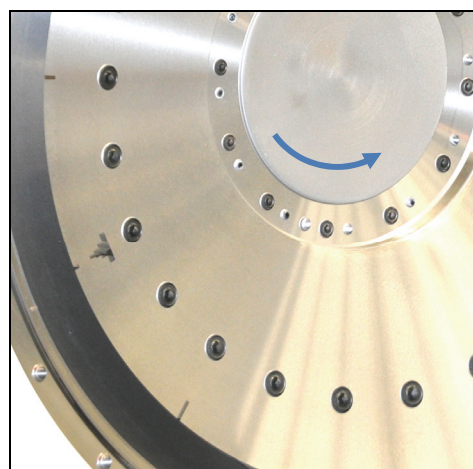


FIGURE 4. 416 stainless steel chopper disk with multiple-width slots. The chopper is positioned relative to the beam with a sub-slide to use different radii of the disk for chopping.

optics improves thermal drift of the mirror alignment and allows continuous operation of the high speed chopper without overheating. The heat is carried away by cooling water continuously pumped through the rotating disk. The temperature rise of the coolant is 2°C due to 520 W beam with a flow rate of 4 L/min. In order to maintain this flow rate at 5,000 RPM, a pump pressure of 240 kPa (35 PSI) is required. The cooling water is supplied to the disk through the non-contact fluid rotary union shown in Figure 5. Air bearings are used on either side of the inlet and outlet to prevent leakage without introducing unwanted error motion, jitter, and wear—critical for a component expected to log over 5,000 hours per year.

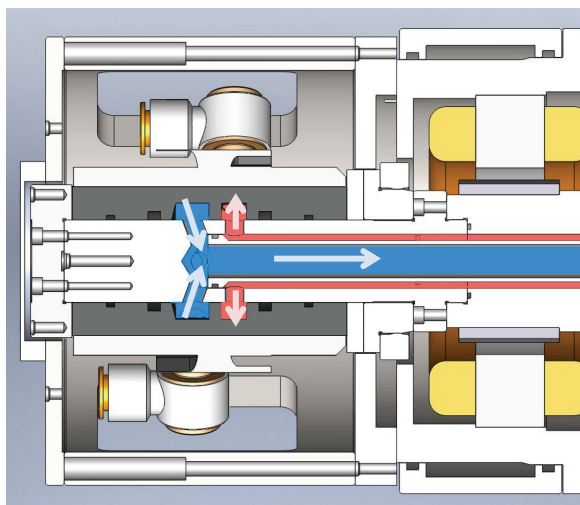


FIGURE 5. Air bearing rotary union for the coolant inlet and outlet. The cold coolant (blue) enters the center and travels toward the disk indicated by the arrow. Return coolant (red) flows back along the outside of the central tube.

Owing to their low error motions and torque ripple, precision air bearing spindles are the ideal choice for low-jitter choppers. The air bearing spindle is a captured thrust, groove compensated design with a vacuum chamber integral to the housing (Model ISO 4.5, Professional Instruments Company). A frameless brushless servo motor (MCS) and encoder (Model ERM 280, Heidenhain) is mounted directly to the rotor of the air bearing spindle. The Heidenhain encoder has 600 lines but it is 2X interpolated for the 1200 line count required by BioCARS. (Encoders with a 1200 line count are relatively unusual and would typically only be available on a small kit encoder.)

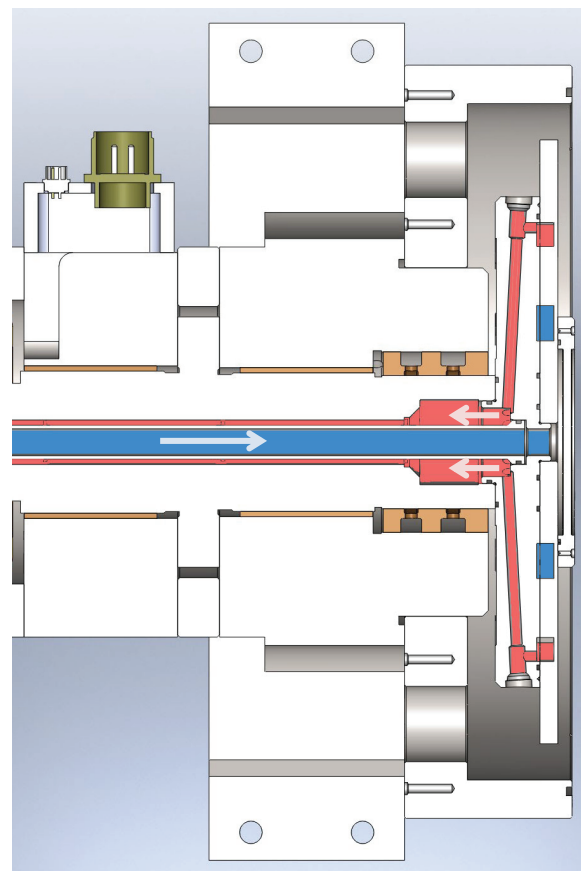


FIGURE 6. Coolant delivery path to the rotating disk, through the cooling passages in the disk, and back to the fluid rotary union.

The ERM would not be considered a high accuracy encoder, but it can be centered to reduce first harmonic errors.

The spindle speed is controlled with a linear amplifier using Frequency Lock Velocity Control (Model LA2000, MCS). The reference signal for the control is supplied by the storage-ring reference clock enabling synchronization with the high speed chopper.

JITTER TESTING

Speed stability is evaluated by measuring the time required for one revolution of the spindle as shown by the index pulse of the rotary encoder. An HP 33120A Function Generator is used to simulate the storage-ring reference signal while recording the speed stability with the HP 5371A Time and Frequency Analyzer (150 picosecond RMS single shot resolution). The internal reference oscillator of the HP 5371A (10 MHz with 5 parts in 10^{10} per day stability) is used as the master clock for both instruments.



FIGURE 7. Jitter measurement setup. An HP 33120A Function Generator is used as the reference signal to the linear amplifier with Frequency Lock Velocity control while speed stability is recorded by the HP 5371A Time and Frequency Analyzer via the period of the encoder index signal.

The vacuum chamber is evacuated to a partial vacuum of approximately 10-15 Torr during jitter testing. In addition, coolant was pumped through the chopper at a flow rate of 4 L/min to include any additional jitter caused by the flow.

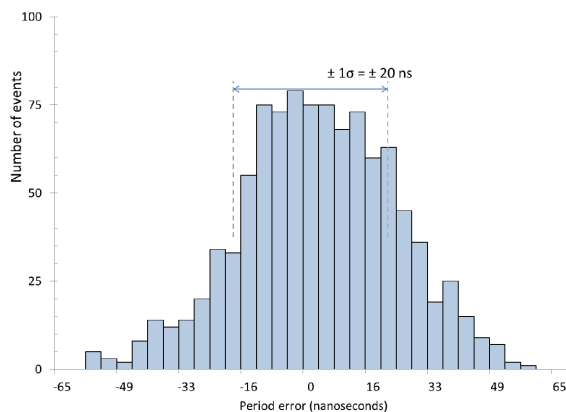


FIGURE 8. Histogram of the jitter measurement with 4 LPM water flow rate and spindle speed of 5,000 RPM. The standard deviation of the 1,000-point sample is $\sigma = 20 \text{ ns}$.

The results of the test demonstrate speed stability better than 20 nanoseconds one σ at a spindle speed of 5,000 RPM as shown in Figure 8. The jitter measurement revealed that the standard deviation of an 1,000 point sample did not exceed 25 ns at 4,000, 5,000, and 6,000 RPM. In fact, the jitter at each speed did not exceed a standard deviation of 2 parts per million relative to the period. For comparison,

published results for a water-cooled heat load chopper using rolling-element bearings and ferrofluidic seals exhibits 2 μs jitter at 6,000 RPM [3].

CONCLUSIONS

Design considerations and test results for a water-cooled air bearing heat load chopper with integral vacuum chamber are described. This chopper was recently installed at BioCARS Sector 14 where it has been operating at vacuum levels as low as 5×10^{-6} Torr. The heat load chopper is necessary as it protects downstream equipment and samples from damage from an intense X-ray beam. The heat is carried away by coolant that is continuously pumped through the rotating disk. The jitter measurement demonstrated 2 parts per million speed stability, or σ less than 25 ns at speeds of 4,000, 5,000, and 6,000 RPM.

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AIR-BEARING FOR NEXT GENERATION CT SCANNERS

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ABSTRACT

The current Computed Tomography (CT) rotor ball bearing architecture is on the edge of its capabilities for the most demanding applications (high rotational speed). Therefore an air-bearing system is designed which is suitable for coming generations. In this article it is described why and what type of air-bearings are chosen. The air bearing configuration and design is explained. Also shape and material choice of the running surfaces (race) of the air-bearings and connection of the race to the scanner rotor is described. Finally test results are presented

INTRODUCTION

Philips is one of the biggest manufacturers of CT scanners. Development and production are located in Cleveland. The first CT scan dates back to 1972. The scan took 10 minutes and a one-slice picture was made. During the following decennia developments resulted in scans with multiple slices, high rotational speeds and source power.

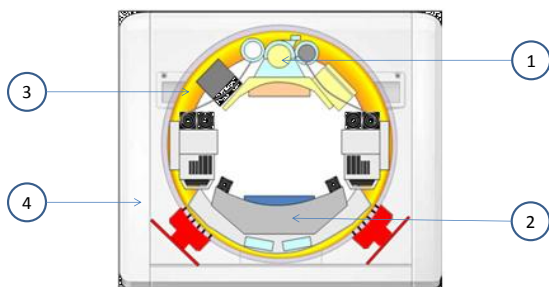


Figure 1 main components of a CT scanner

A CT scanner consists of a source (1) (see Figure 1) which x-rays pass through a patient and hit a two dimensional detector (2). The detector converts the x-rays into electrical signals with which an image is constructed. The x-ray source and detector are mounted on a rotor (3) which revolves around the patient in order to get a three dimensional picture. The mechanics on which the source and detector are mounted is called the gantry (4). The assignment for Philips CT Cleveland and Philips Innovation Services (PInS) was to jointly

develop a novel advanced gantry platform (frame, rotor structure plus bearing), which can be used for the coming generations. This article focuses on the design of the bearing system.

REQUIREMENTS

To be able to make a still picture of a heart (0.2 sec between two beats) the rotor has to rotate with a velocity of 300 rpm. At the bearing radius the speed will be up to 30 m/s. With this speed and weight of the rotor (approx. 1200 kg) the kinetic energy (260 kJ) is comparable to a midsize car at a 75 km/h speed. The scanner must be designed to resist an over-speed test up to 450 rpm, and must have emergency running properties. To ensure image quality, rotor suspension frequencies need to be higher than 20-25 Hz and rotor run-out smaller than 0.1 mm. Also static imbalance should be below 0.3 kgm, and the dynamic imbalance should be smaller than 0.08 kgm² to not excite the rotor-dynamics. The rotor must be up to speed within 30 sec.

SYSTEM DESCRIPTION

The roller bearing architecture used in today's CT scanners is reaching its performance and reliability limits. Air bearing technology seems the appropriate technology to meet the requirements that apply to future CT generations of scanners

Bearing System

A high speed air bearing motion platform can operate up to and beyond 450 rpm. Static and moving parts have no contact and therefore no wear which results in a reliable system with extended system life time and low audible noise. Because the air bearing system works with small air gaps (< 10 µm), bearing and running surface accuracies have to be in the µm regime; this results in an accurate system directly benefiting quality of imaging system.

Given the large diameter (approx. 1.6 m) the air bearing concept must be based on air-bearing pads which run on an accurate ring (race). The configuration which is chosen consists of two

radial bearings pads carrying the weight of the rotor (gravity preloaded). Furthermore the rotor is kept in position by three fixed air-bearing pads under an angle of 120° (see Figure 2). The two upper axial air-bearings are preloaded by spring loaded opposing air-bearing pads. The center of gravity has an offset with respect to the bearing plane. As result the lower axial air-bearing can be gravity preloaded.

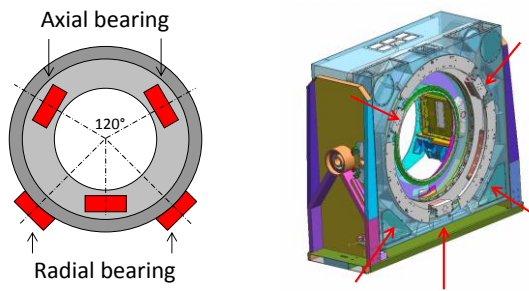


Figure 2 Bearing configuration

Drive System

The gantry is driven by an induction motor. This type of motor does not have magnets which may come loose because of the high centrifugal forces. The induction motor is divided into three segments which are place between the air-bearings pads, also under an angle of 120° (see Figure 3). Besides a driving force the motor segments apply (parasitic) attraction forces to the rotor. The advantage of this configuration is that the driving forces exert a pure torque and parasitic forces cancel each other out and as a result will not load the air-bearing system.

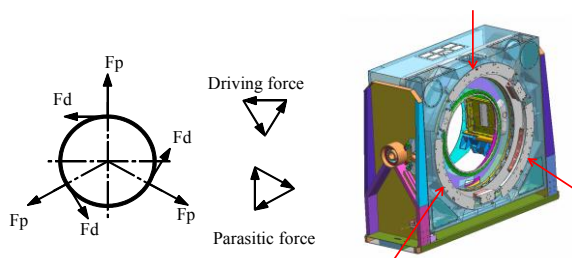


Figure 3 Forces motor configuration

Air-Bearings

The air-bearing pads are placed on a large radius which results in a high circumferential velocity. This high velocity causes significant drag and a shifted pressure distribution, which in turn affect the load-ability. This is especially true for orifice bearings (see Figure 4). In contrast to orifice bearings; porous air-bearings have high stiffness and load-ability also at higher speed,

low air usage and low audible noise. The porous material is graphite which also has good emergency running properties.

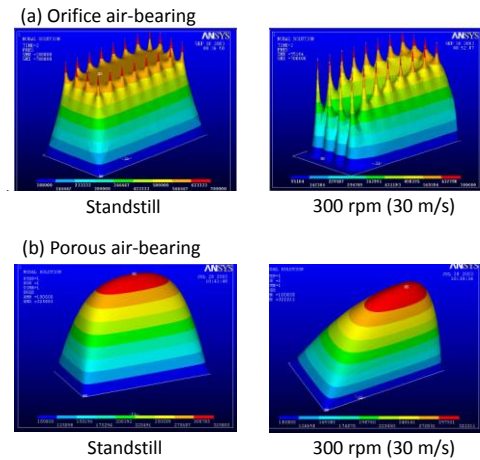


Figure 4 pressure distribution air-bearings at different velocities

Air-Bearing Support

A CT scanner should work on low and high velocities. Because of the pressure distribution shift, the pivots of the air-bearings are not positioned under the center of the bearing but with an offset (see Figure 5). This offset keeps the bearing gap parallel at the maximum speed of 30 m/s. At standstill and lower speeds the bearing will have a bearing gap that is largest at the bearing front and smallest at the trailing edge.

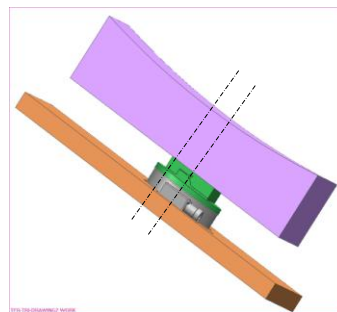


Figure 5 Position of air-bearing support under the air bearing

To obtain the right dynamic behavior and CT system accuracy, the rotor's lowest eigen frequency should be higher than 20-25 Hz. For the whole mechanics, rotor and gantry inclusive of bearing and bearing support, this requires a minimal suspension stiffness $1e7$ N/m. For the air-bearing support a stiffness of $1e8$ N/m is budgeted. Besides the supporting function, the air-bearing support must allow the air-bearing to

align itself to the running surface of the race. This means a low stiffness in two tilt directions (R_x and R_y , see Figure 6). To prevent vibration of the air-bearing the third tilt direction (R_z) should be supported in a stiff way. By making use of elastic elements the ratio between stiff and compliant directions can be made very large. Because the flexible hinges are short they are very stiff in the perpendicular (x -, y -) direction, without losing compliance in the swiveling direction. Also the narrow gaps as a side effect of manufacturing (EDM) prevent the flexures from being overloaded.

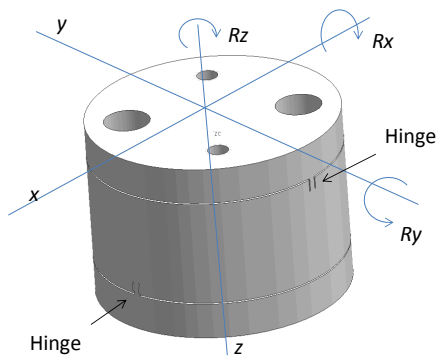


Figure 6 Air-bearing support with flexible elements

Between the two hinge lines (see Figure 6) which are at 90° angle sufficient length is needed to obtain a stiff mid-section for force transfer from one hinge to the other.

Race

An accurate ring (race) with a surface roughness of $R_a = 0.4 \mu\text{m}$ is designed as running surface, for the air bearing. The surface tolerance over the bearing length had to be within $2 \mu\text{m}$. The air-bearings are positioned on the outside of the rotor for packaging reasons. The cross section of the race is T-shaped (see Figure 7). In this way a stable and light weight race is obtained.

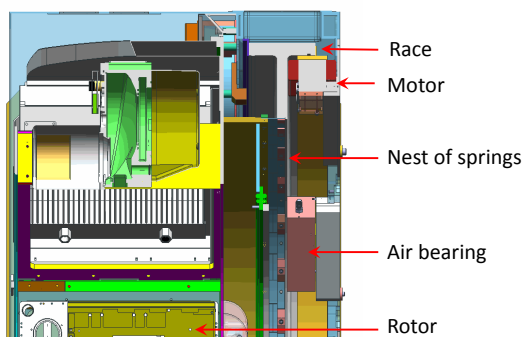


Figure 7 cross section of CT scanner

In cases of emergency, when the air-pressure fails, the race with air-bearings is dry running. Also during normal use the race will at times contact the air-bearing. To come to the right material combination several tests have been done. A Stavax race with a low surface roughness appeared to be having the best, most robust running surface for a porous air-bearing. A compromise between robustness and corrosion resistance on one hand and stable material on the other hand had to be attained. A stable material is a material with no internal stresses. Internal stresses deform material and therefore shape of parts, during life time. Internal stresses can be cancelled by proper heat treatment. This affects the hardness and therefore the robustness and corrosion resistance. The compromise is chosen on 40 HRC

Rotor - Race Connection

The race and rotor need to be connected to each other well aligned. Because of mounting and centrifugal forces and temperature changes the rotor will elastically deform. Using a nest of springs [1] deformations will not affect the race shape. The implementation of the nest of springs is a monolithic part with H-shaped cut-outs (see Figure 8). The parts between the cut-outs are leaf-springs with low radial and high tangential stiffness.

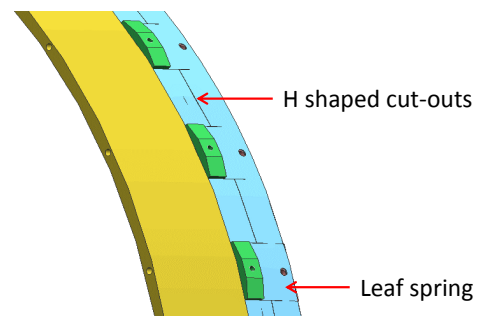


Figure 8 Monolithic nest of springs

MEASUREMENT

The running accuracy of the CT rotor is measured using the spindle error analyzer of IBS Precision Engineering (www.ibspe.com). Capacitive gages measure the movements of a calibrated precision ball, which is mounted on the rotor of the CT scanner (see Figure 9). The measurements are performed at different rotation speeds: 30, 60, 120, 180 and 220 rpm.

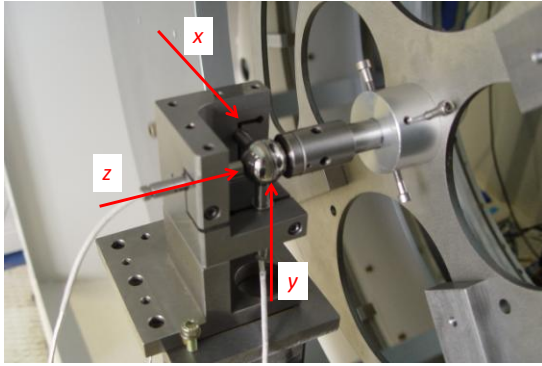


Figure 9 Rotor accuracy measurement setup

The signals of x and y are combined and transformed to a rotating coordinate system. The following deviations are measured: for 30 rpm speed deviation is $21\text{ }\mu\text{m}$ and for 220 rpm the deviation is $64\text{ }\mu\text{m}$, which is sufficiently accurate for the required image quality

As previously stated the system contains a large amount of kinetic energy (260kJ) which could lead to dangerous situations when the system is not undoubtedly safe. In particular the situation where air falls off is considered very risky. Therefore this situation has been simulated in a test by shutting the air off when the gantry is at full speed. This has been done 20 times in a row without servicing the air-bearings. After each “crash” the system could be restarted without problems. This confirmed that the porous air-bearing material has excellent emergency running properties and therefore is safe to use in a CT scanner application.

CONCLUSION

The proposed air-bearing architecture together with the induction motor drive is capable of 300 rpm within $64\text{ }\mu\text{m}$ accuracy at a 20 Hz suspension frequency. The air bearing system is part of the current Philips Brilliance iCT (intelligent Computer Tomography) diagnostic imaging system, which is among the best performing CT scanners of this moment



Figure 10 Press reviews

ACKNOWLEDGEMENTS

Measurements were done together with IBS which was also the representative from New Way Air bearings. Thanks the CT development team in Cleveland, especially to Ron Sharpless, for their great contribution.

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OPTIMISING MACHINE PERFORMANCE BY NOVEL DESIGN

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OVERVIEW

There is a need to improve machine tool performance, accuracy, efficiency and reliability whilst minimising cost. Classical machine tool designs are prone to axis displacement errors that are governed by the principle of the six degrees of freedom of a rigid body. Often to achieve specification, these errors must be minimised by methods such as precision machining, scraping, and the addition of complex metrology, software compensation and precise thermal control. This is time consuming and expensive. It can lead to added complexity and thereby reduce reliability. The machine described here as an example is a large diamond turning lathe used in the production of optical quality micro structured surfaces. These surfaces require many hours to machine and stability over time, accuracy and reliability are paramount.

Machine tools are designed to cover a range of options that have been specified by analysis of market requirements. In reality a user may never need many of the options the supplier has provided and for which the end user has paid. Careful analysis of user needs can focus the design onto the key requirements.

This paper describes how a novel design can virtually eliminate the errors associated with conventional machine designs. Error budgets, analysis of the key performance drivers of the machine and the products it is to produce are used to identify which elements require the most attention.

INTRODUCTION

Optical quality micro structured films are a key element in the flat panel displays we all use at work and at home. Their manufacture requires tooling of the very highest quality. More often this is in the form of tooling rolls which are diamond turned or fly cut on roll turning machines. Precision, better than 1 in 1×10^{-6} is required on components with diameters of up to 500mm and face lengths of 2000mm. Achieving accuracies at this level requires machine tool technology that is truly 'state of the art'.

BACKGROUND

Traditional machine tools generally conform to standard design types that have been refined over years of development. They are made to cover a broad range of requirements and therefore are a compromise between ultimate performance, capacity and cost. For example a lathe will be designed to machine parts of a certain size (diameter and length). This is accomplished by stacking an X axis (perpendicular to the workpiece axis) on a Z axis (parallel to the workpiece axis) which allows the tool to pass anywhere in the specified diameter and length envelope. Necessity requires the axis guideways to be placed at a compromise position to allow the structure to be sufficiently stiff and accurate. Because of this the Z axis guideways are well below the work piece and tool centre line.

Sources of Error.

The offset from the guideway to the work piece and tool centre line exaggerates the inevitable manufacturing (straightness) errors in the guideways and causes a roll error (X displacement) as the axis moves in the Z direction. Z axis roll errors are a primary source of component error and their effect is to produce parts with a cylindricity error. This may not be a problem for general machine shop lathes where tolerances of 3-5 microns may be acceptable but for the ultimate precision required for tooling rolls, sub-micron performance is required.

Improving Performance.

Improving performance requires careful analysis of the sources of error and a useful tool to achieve this is the 'error budget'. (See Fig. 4). The error budget identifies the problem areas and focuses attention on the key error drivers. Roll offset is one of the key drivers of error in any lathe. The error budget will compare the difference in predicted performance for conventional and novel machine designs.

Error correction can be used to reduce errors. This can be done by either an open loop or closed loop method.

Open loop is where an axis is calibrated during build and a correction table is created and used to alter the axis position as it moves. (This can only be done if a suitable axis exists – for a lathe the X axis can compensate for displacement errors in the X direction caused by the roll of the Z carriage as it moves in the Z direction). Occasional recalibration may be required to maintain an acceptable level of accuracy. Closed loop is where axis error is constantly measured against a known standard during operation and corrected by a compensating axis. This option may give the best results but requires significant investment in on-machine metrology, usually by laser, straight edge or a combination of both.

For any machine tool, errors need to be measured close to the tool position which often, because of space limitations, leads to a compromise between machine operation and the need for on machine metrology.

REMOVING ERROR BY NOVEL DESIGN

Performance and cost can be significantly improved by considering the machine's true purpose and designing it for that purpose only. The example here is a lathe for turning cylinders (See Fig. 9). The aim is to machine cylinders within a range of diameter and length. Most of the turning is done by 'fixing' the X position and for the tool to be driven along the work piece by the Z axis to produce a cylinder. Small movements of the X axis produce the structured surface required.

The aim therefore should be to design with that as the primary focus of attention and not to be distracted from that aim without good cause. Previously the error budget has highlighted that a major source of intrinsic error is caused by the Z axis roll and that the roll offset magnifies this error. However conventional lathe designs require the Z axis guideways to be placed well below the tool position at the work piece center to facilitate access to a wide range of component types. If now, considering the axis strokes needed to achieve the actual range of work piece sizes required, the Z axis guideways are moved so the center of roll is at the center of the work piece the roll error disappears. (See Fig. 1). It should be noted here that roll error motion does not take place about any particular center. This is because roll, in general, is a product of straightness or translation errors.

However, it is reasonable to assume, from the point of comparison, that it's origin is midway between the axis guideways.

This novel, vertical offset guideway bed design not only reduces roll error but increases stiffness by around 1.6 times, validating the error budget regardless of part size and mass.

In reality it is more likely that a compromise will have to be made to facilitate the addition of a B axis and tool post, etc. but because the true purpose "to machine a range of precision cylinders" has been put at the forefront of the design strategy, a design can be produced that significantly reduces the major machine intrinsic errors without the need for complex metrology. This strategy can be used on other axes until a novel machine design is produced that not only meets the requirements but also simplifies the design leading to better performance, reliability and lower cost.

MAKING THE WRONG ASSUMPTION

Although at first it may appear that the same concept may work for a similar process in reality the opposite may be true. An example of this came to light when a project to revolutionise the process to produce precision gravure printing cylinders was proposed. This required two machines. One to finish machine the cylinder and the other to engrave a pattern on the surface. Machining the cylindrical surface required a high precision machine that produced a constant diameter with a high degree of surface finish (25-50 nanometers Ra) and the lathe concept described here was considered to be ideal for this requirement

The engraving process required a laser to ablate the surface in series of pits accurately positioned around the circumference. The laser has a long focal length so errors in the X direction that had been critical for accurate roll turning are now relatively unimportant. However the position of the pits around the circumference is critical and therefore the roll error, that had been designed to coincide with the cylinders center, now produces an error that is most unwelcome.

The answer was to design an engraving machine that had the opposite characteristics to the turning machine (See Fig. 3). The roll center was moved to a position where it produced errors in the X direction and only minimal errors in the Y direction.

This example highlights the importance of analyzing the process for which the machine is being designed. This forces the designers attention toward the critical sources of error and

how to moderate them to produce the best performance while minimising complexity and cost.

ERROR BUDGETS

Error budgets are used as a tool for comparing designs and as a guide to predicting performance [2]. They highlight areas where errors may be considered to be too large for the design to continue without change. They can also be generated from the specification, before any design has commenced, to focus attention onto key areas.

The following simplified layouts and error budgets show a comparison between the novel lathe concept, a conventional lathe design and the engraving machine.

They clearly show the benefits that can be gained by correct positioning of axes relative to the work piece and the production process.

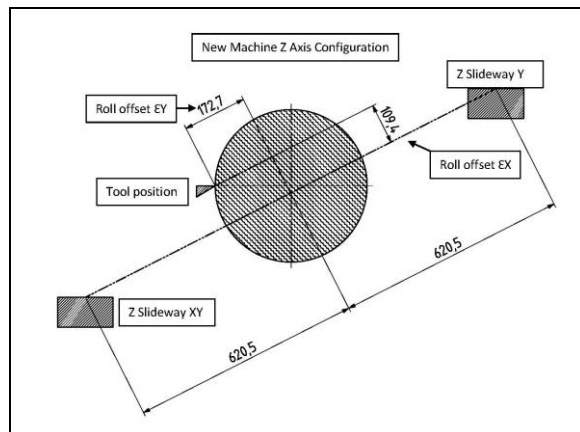


Fig. 1. New Machine Z Axis Concept

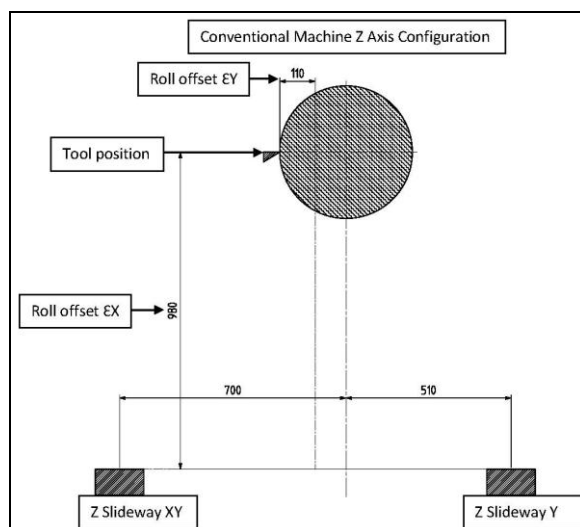


Fig. 2. Original DT Machine Z Axis Concept.

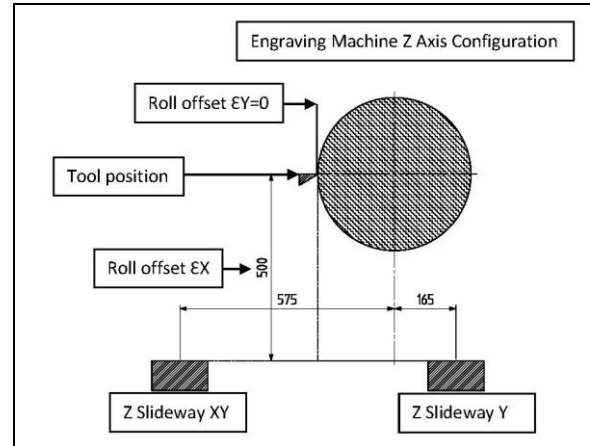


Fig. 3. Engraving Machine Z Axis Concept.

New Machine Concept Simplified X and Y Axis Geometric Error Budget									
Axis	Feature	Units	Intrinsic	Abbe Error	Roll Offset	Max Error (um)			
				(m)	(m)	X	Y	Z	
X Axis	Pitch	Arc Sec	2	0.19		1.90			
	Yaw	Arc Sec	2	0		0.00			
	Roll Error in Y	Arc Sec	2		0		0.00		
	Roll Error in Z	Arc Sec	2		0.19			1.90	
	Straightness-Y	um	0.75				0.75		
	Straightness-Z	um	0.75					0.75	
	Encoder Error	um	2			2.00			
	Thermal Error	C	0.01			0.05			
Z Axis	Pitch	Arc Sec	2	0.594				5.94	
	Yaw	Arc Sec	2	0.336				3.36	
	Roll Error in X	Arc Sec	2		0.109	1.09			
	Roll Error in Y	Arc Sec	2		0.173		1.73		
	Straightness-Y	um	3				3.00		
	Straightness-X	um	3					3.00	
	Encoder Error	um	2			3.00			
	Thermal Error	C	0.01			0.05			
Total Error (um)						8.04	5.48	14.00	
RSS Error (um)						4.22	3.54	7.40	

Fig. 4. Error budget New Machine.

Conventional Machine Concept Simplified X and Y Axis Geometric Error Budget									
Axis	Feature	Units	Intrinsic	Abbe Error	Roll Offset	Max Error (um)			
				(m)	(m)	X	Y	Z	
X Axis	Pitch	Arc Sec	2	0.544		5.44			
	Yaw	Arc Sec	2	0		0.00			
	Roll Error in Y	Arc Sec	2		0		0.00		
	Roll Error in Z	Arc Sec	2		0.544			5.44	
	Straightness-Y	um	0.75				0.75		
	Straightness-Z	um	0.75					0.75	
	Encoder Error	um	2			2.00			
	Thermal Error	C	0.01			0.05			
Z Axis	Pitch	Arc Sec	2	0.594				5.94	
	Yaw	Arc Sec	2	0.336				3.36	
	Roll Error in X	Arc Sec	2		0.98	9.80			
	Roll Error in Y	Arc Sec	2		0.11		1.10		
	Straightness-Y	um	3				3.00		
	Straightness-X	um	3					3.00	
	Encoder Error	um	2			3.00			
	Thermal Error	C	0.01			0.05			
Total Error (um)						20.29	4.85	17.54	
RSS Error (um)						11.77	3.28	8.99	

Fig. 5. Error budget for Conventional Machine.

Engraving Machine Concept Simplified X and Y Axis Geometric Error Budget									
Axis	Feature	Units	Intrinsic	Abbe Error	Roll Offset	Max Error (um)			
				(m)	(m)	X	Y	Z	
X Axis	Pitch	Arc Sec	2	0.2		2.00			
	Yaw	Arc Sec	2	0		0.00			
	Roll Error in Y	Arc Sec	2		0		0.00		
	Roll Error in Z	Arc Sec	2		0.2			2.00	
	Straightness-Y	um	0.75				0.75		
	Straightness-Z	um	0.75					0.75	
	Encoder Error	um	2			2.00			
	Thermal Error	C	0.01			0.05			
Z Axis	Pitch	Arc Sec	2	0.594				5.94	
	Yaw	Arc Sec	2	0.336				3.36	
	Roll Error in X	Arc Sec	2		0.5	5.00			
	Roll Error in Y	Arc Sec	2		0		0.00		
	Straightness-Y	um	3				3.00		
	Straightness-X	um	3					3.00	
	Encoder Error	um	2			3.00			
	Thermal Error	C	0.01			0.05			
Total Error (um)						12.05	3.75	14.10	
RSS Error (um)						6.48	3.09	7.43	

Fig. 6. Error budget for Engraving Machine.

BED DESIGN

The hydrostatic Z guideway configuration required for this concept required an unusual bed design. A number of important considerations were included in the bed design process. These included:

1. A stable, self-stiff, well damped machine bed that did not need a specially constructed isolated foundation,
2. An acceptable working height without the need for a separate mezzanine structure,
3. Simple, commercially available construction technique.

The offset bed design lends itself to all these requirements and a thin walled fabrication filled with a granite based resin composite produced the desired result. Cast iron bars secured to the fabrication were part of the composite bed and these provided the interface for the guideways, linear motors and encoding system. Fabricated bulkheads and cross members allowed the bed to be tuned to the desired target performance. Finite element analysis was used extensively to optimize the design including varying the work piece length, mass and the tailstock position to simulate changing operating conditions. Results were compared to verify that the changes in deflection and alignment of headstock, tailstock and Z carriage were within acceptable limits.

The bed, along with the rest of the machine's structure and axes, had to be temperature controlled to ensure stability over extended machining periods. This required an oil shower controlled to 0,01° Celsius.

SPINDLE DESIGN

To produce optical quality surfaces it is essential that the spindles, used to rotate the workpiece, have excellent synchronous and asynchronous error motions.

The development of the spindle design included ensuring that the spindle shaft could be ground without needing any component or grinding set up changes. This resulted in a journal with very low roundness errors ($<0,2\mu\text{m}$) and no noticeable lobing. Together with a five pocket hydrostatic journal design, this achieved excellent radial error motions

C Axis Spindle Error Motions @ 700revmin⁻¹		
	Synchronous	Asynchronous
Radial Spec.	0,125 μm	0,075 μm
Radial Actual	0,070μm	0,070μm
Axial Spec.	0,100 μm	0,075 μm
Axial Actual	0,020μm	0,060μm

Fig. 7. C Spindle Error Motions.

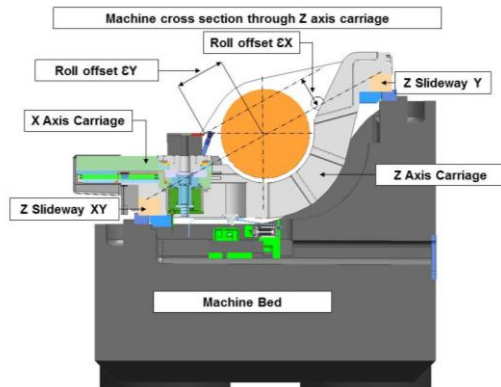


Fig. 8. Machine Cross section



Fig. 9. Completed Machine.

SUMMARY

Machine tools are the backbone of all manufacturing processes. Reducing their complexity, improving performance, reliability and cost effectiveness should be the aim of all new machine designs. Achieving this goal requires the customer, designer and manufacturer to consider the machines true purpose and to focus the design to achieve the best possible result in a cost effective manner.

This design has achieved this by removing the need for complex closed loop error correction. The offset bed and novel carriage has increased stiffness and reduced the sensitivity to roll errors over a wide range of components.

ACKNOWLEDGEMENTS

Thanks to Mike Clarke, Trevor Joyce, Colin Knowles, Mike Pierse, Mark Stocker and Lorena Lopez-Pinana for their contribution.

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VISCOUS SHEAR IN AIR BEARING GAPS FOR PRECISE WEB TENSION AND TEMPERATURE CONTROL

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INTRODUCTION

Air bearings have enabled the precision and speed required in motion control systems for manufacturing advanced IC circuits. But today the substrates for manufacturing circuits are changing. Flat-panel display glass substrates can be meters square, substrates for solar applications and printed electronics on roll-to-roll substrates are demanding completely new precision motion systems. Air bearings are again providing breakthrough, enabling technology by acting directly on the substrates. So flat-panel glasses are controlled through processes by air bearings without contact rather than being vacuumed to a Chuck. This research explores flexible films flowing over "Air Turns" rather than contact rollers and the use of viscous shear in the air films for motion control, cleaning, drying and precision temperature control. This would help satisfy demands from the printed electronics industry for non contact precision manufacturing of Flexible Displays, OLED Lighting, PV, Smart Packaging and Printed Batteries.

SHEAR FORCES TO CONTROL TENSION

An air turn is an air bearing based, usually non-rotating roller which flexible films would float over and around that would replace contact rollers. The elimination of contact roller friction means uniform tension between many air turns and the possibility of much better control of web tension.

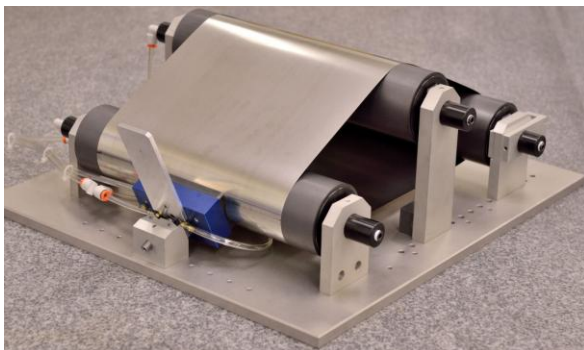


FIGURE 1.

By sandwiching the web between two air bearing films, high pressures may be exerted on the film in compression. Air shear in these air bearing gaps, acts directly on the flexible film for: driving the flexible film, controlling film tension between rollers, creating a spreading tension across the web (along the length of a roller) and centering the film on the roller.

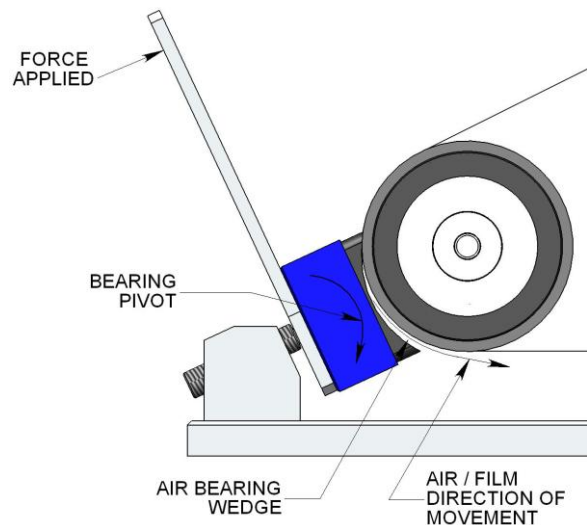


FIGURE 2.

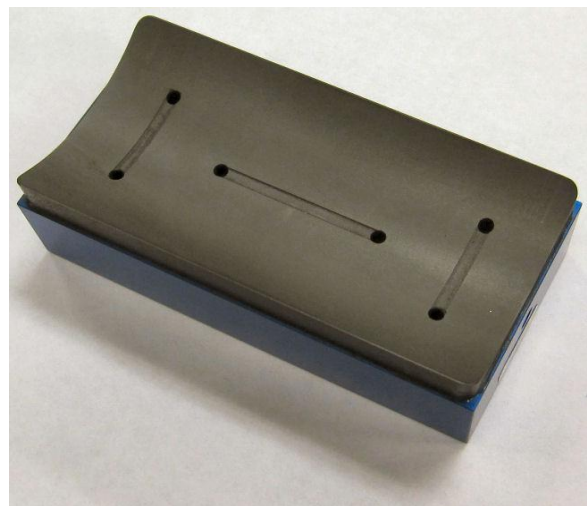


FIGURE 3.

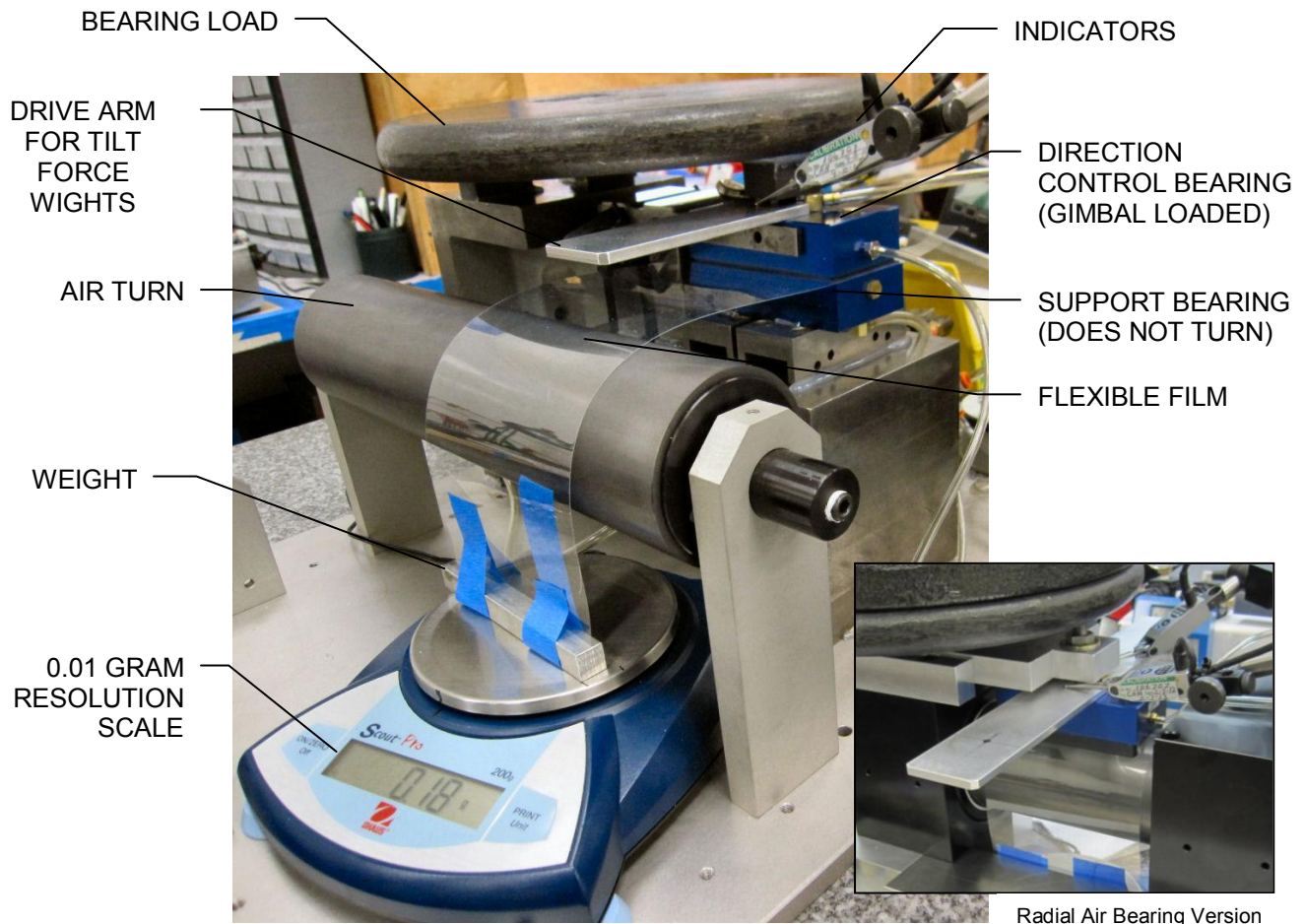


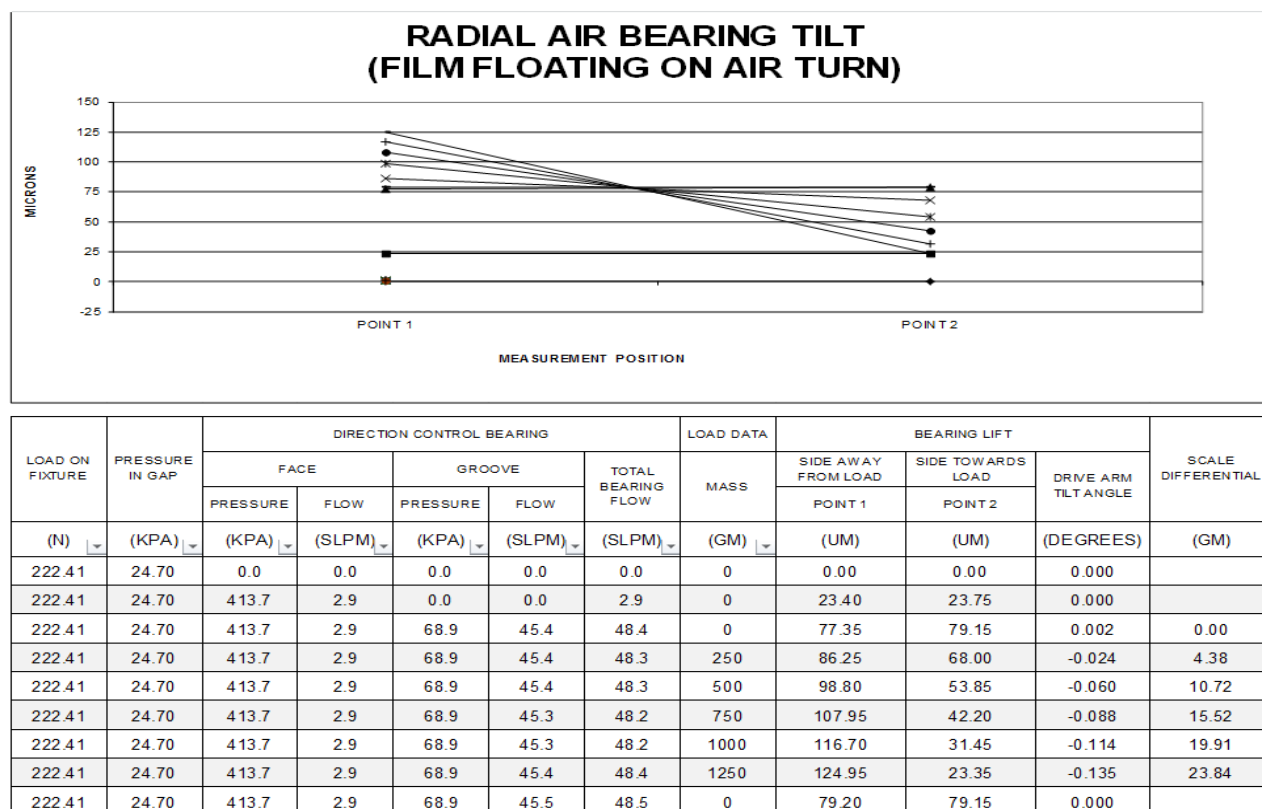
FIGURE 4.

TEST SET UP

A test fixture (Figure 4) was built to measure the pull force exerted on a piece of film under varying test conditions: Load, air pressure, air flow, and tapers in the air gaps. A weight was attached to one end of a piece of flexible film and the film was draped over a frictionless New Way Air Turn air bearing. A precision scale was positioned so that the weight was resting on top of the platter of the scale. The other end of the film was sandwiched between a pair of air bearings. The lower bearing was a stationary support bearing that was affixed to the test fixture to ensure that it did not move. The upper bearing, loaded by point contact, floated parallel to the support bearing on a cushion of air and had angular freedom of movement, electronic indicators were positioned at each edge. A set of weights were positioned on top of the fixture in a three point stance with one of the positions

being the tooling ball that was located in the center of the upper bearing. This weight provided a downward force on the direction-control bearing and squeezed the air film to increase the pressure in the air gaps between the bearing faces and the film. A drive arm was attached to the back of the direction-control bearing, and weights applied to the ends of the drive arm would cause the bearing to tilt slightly out of parallel with the support bearing which would cause the air in the gap to move towards the larger opening at the opposite side from the load. The air shear from this movement would pull on the surface of the film and create a force which could be measured as a change in the weight measurement displayed by the scale. This set up was reconfigured to also test radiused bearings on an Air Turn. (See inset.)

TABLE 1.



TEST CONDITIONS

The data in table was taken using the New Way Porous Media Radial Air Bearing shown in figure 3 which has grooves and feed/exhaust holes for experimental purposes. In this case additional air pressure was fed into both the holes in the single center groove. This groove is directly under the point loading. The air fed into this groove was a 20 times multiple of the air feeding the face of the bearing even at one third the pressure, yet the total flow of 48 SLPM is still reasonable. This test was repeated many times varying the dead weight load on the bearing, air pressure to the grooves and groove configurations. Higher loading contributed to similar forces at lower air flows. Tilt in the gap was the dominant variable and interestingly resulted in a very linear plots.

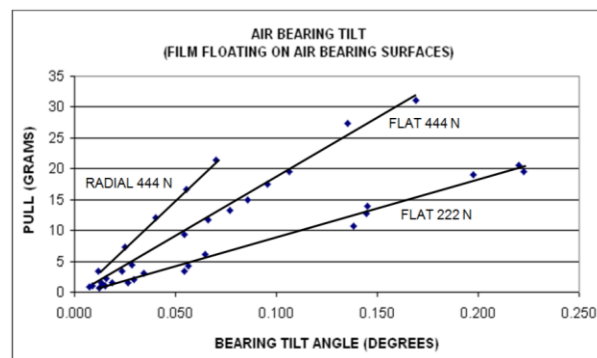


FIGURE 5.

RESULTS SUMMARY

The results of this testing showed that a single 50 x 100 mm air bearing could generate over 30 grams of shear force on the flexible film across the 50mm direction. And this force could be controlled with a resolution to 100th of a gram. So here we have a technique that can control tension between rollers in a continuous line, and can be used to create a shear force across the roller essentially taking wrinkles out, and potentially being able to stretch and align a pattern on a web without contact with either side of the web.

TEMPERATURE CONTROL

Additionally, this technique can be very effective at driving the temperature of the web by acting on both sides of the web simultaneously. Because of the high relative pressure (1 to 3 atmospheres) and such high velocities of air in the air gap the boundary layer of air associated with the surface of the film can be scrubbed. This is essentially increasing convective film coefficients (wind chill) and so more effectively driving the temperature of the web than heated or cooled rollers that are in contact with the web. There are two reasons for this: First, there isn't a lot of contact between a film and a roller, as both have surface roughness that greatly minimizes actual contact between the film and roller; and second, it leaves an insulating air layer. As the roller and the film are driven at the same speed, this layer is stationary relative to the roller and the film. At higher speeds more air is entrained through dynamic effects so the layer gets thicker but is still static with respect to the roller and film.

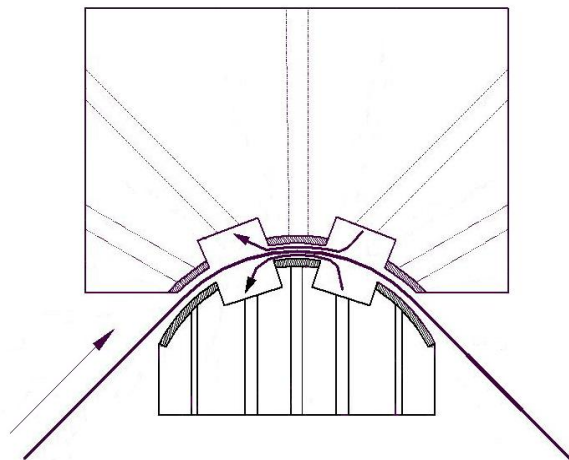


FIGURE 6.

OTHER FUNCTIONALITIES

This same shear force or boundary layer scrubbing can also be very effective for cleaning and conditioning films before other processes. This technique may also dramatically reduce drying times for the same reasons it is very effective at transferring temperature. The illustration makes the point that cleaning and drying can happen in immediately sequential and contained processes, dramatically reducing the space used in cleaning and drying today.

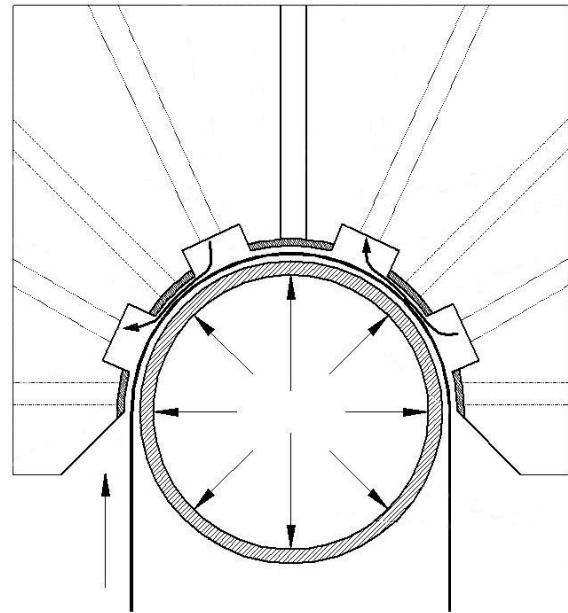


FIGURE 7.

CLOSING COMMENTS

It is recognized that air knives and air bars for turning and drying webs are not new. What New Way is now offering is higher pressure/lower flow web handling components that create much higher gas velocities relative to the film surface. The smaller gaps associated with this technology also reduce flows, and increase the precision with which the web may be handled.

REFERENCE

[1] United States Patent US 8,123,868 B2 Devitt

IMPROVEMENT OF HEAT TRANSFER COEFFICIENT OF BOILING SURFACE ON NANOSTRUCTURED METAL USING NANOIMPRINTING

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1. INTRODUCTION

The clarification of the mechanism of boiling heat transfer or the improvement in heat transfer coefficient is important for enhancing the performance of heat exchangers in various devices, such as air conditioners and chillers for machines.

As shown in Figure 1, the ideal boiling phenomena are as follows: the formation of many nucleation points on the boiling surface and the flow of small bubbles without coalescing with each other. If large bubbles remain on the surface, a heat insulation layer is formed. Here, a critical heat flux, the thermal limit of a phenomenon where a phase change occurs during heating, is one of the indexes for the degree of heat transfer [1][2].

Anderson and Mudawar [3] manufactured cavities mechanically. The mouth diameter of the cavities was 300 μm . They found that artificial cavities had little effect on boiling heat transfer because the cavities were too large to maintain a stable vapor nucleation. Honda et al. [4] studied the effect of square micro-pin fins on a boiling surface. The largest heat flux, about 2.3 times larger than that of a smooth surface, was obtained

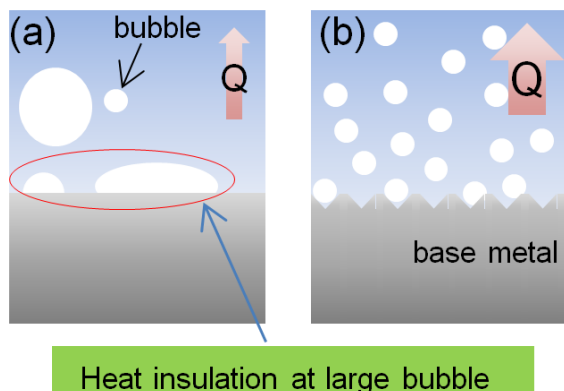


FIGURE 1. Scheme of generation of bubbles on (a) flat surface and (b) nanostructured surface.

when the dimensions of the micro-pin fins were $30 \times 30 \times 60 \mu\text{m}^3$ (width $\times$ depth $\times$ height) and the fin pitch was 60 μm . These results show that the fabrication size has a large effect on the boiling surface, and the effective fabrication size is considered to be less than 300 μm . However, such fine and high-aspect-ratio structures have been fabricated only by lithography and etching. In this paper, we propose a method of fabricating a boiling surface on a metal by nanoimprinting. Nanoimprinting is a method of thermal forming using a nanostructured mold. It is mainly carried out to fabricate polymers or glass. It can fabricate a fine structure in a relatively short time and at a low cost. However, a metal has not yet been fabricated by nanoimprinting; thus, there is a need to verify the possibility of fabricating a metal by nanoimprinting.

2. EXPERIMENTAL

Nanoimprinting

We used electroplated Ni molds with pyramid arrays with pitches of 5 and 20 μm . Figure 2 shows the scanning electron microscopy (SEM) images of a Ni mold. The heights of the pyramids are 2.8 μm (5 μm pitch) and 11 μm (20 μm pitch). The master for Ni electroplating was fabricated by the KOH anisotropic etching of Si.

We used a nanoimprinting machine (SCIVAX VX-1000N-NN, SCIVAX Co., Ltd) and a bulk pure aluminum (Al) plate of 100 μm thickness.

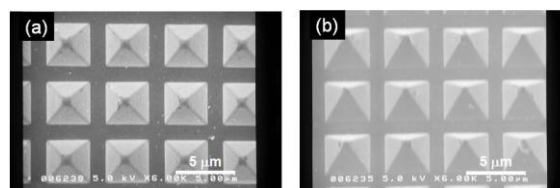


FIGURE 2. SEM images of 5- μm -pitch pyramid array Ni mold: (a) top view and (b) tilt (30°).

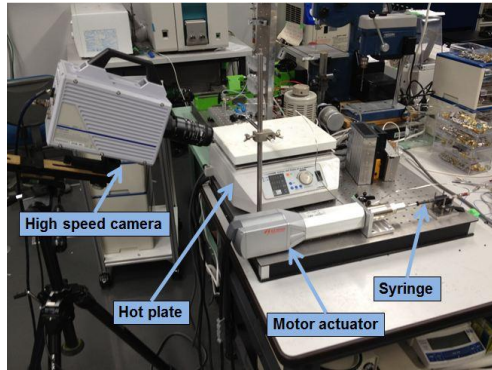


FIGURE 3. Experimental system.

The pressures were verified to be 15 - 450 MPa and the effect of pressure on replication degree was investigated.

Droplet Evaporation

To examine the capability of the boiling surface on bulk Al, we measured the time of droplet evaporation. Figure 3 shows the experimental system used. A droplet of water (26 mg) was placed on the heated Al sample. We made a droplet by using a syringe that can contain 2.5 ml of liquid. The syringe was controlled by a motor actuator (EZHC6A-10, ORIENTAL MOTOR Co., Ltd.). The conditions of the droplet were recorded by using a high-speed camera (FASTCAM SA1.1, Photron Ltd.). We used annealed Al samples with dimensions of $20 \times 20 \times 5 \text{ mm}^3$ (width $\times$ depth $\times$ height) in this experiment to set a thermocouple. The Al samples were polished to a mirror surface before nanoimprinting. The surface roughness of the Al samples after mirror finishing was smaller than 10 nm.

3. RESULTS AND DISCUSSION

Nanoimprinting

Figures 4 and 5 show the SEM images of the surfaces of the nanoimprinted samples. The images show that the pattern of the Ni mold was not imprinted at a pressure of 15 MPa. This is because the pressure between the Ni mold and the Al sample was lower than the yield stress of Al.

On the other hand, the pattern was finely imprinted at the pressures of 45, 90, 220 and 450 MPa. By comparing the figures, it is clear that the imprinted area becomes larger when a higher pressure is applied. We defined the replication degree as the ratio of the diagonal dimension of the imprinted area to the original dimension of the Ni pyramid array mold.

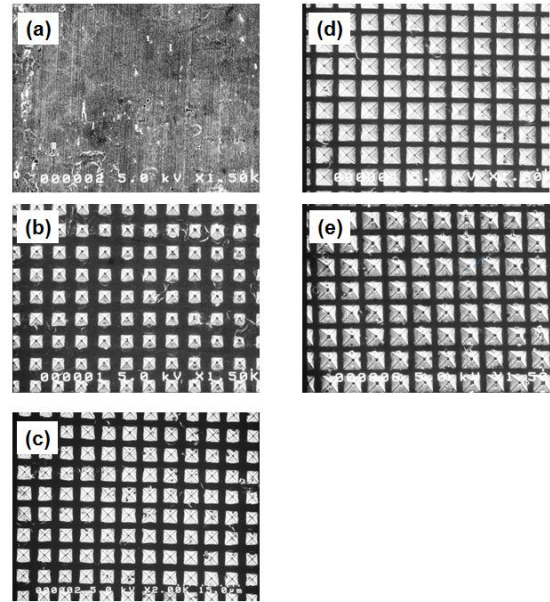


FIGURE 4. SEM images of surfaces of Al samples after nanoimprinting. Pressures between 5- μm -pitch Ni mold and Al samples: (a) 15 MPa (b) 45 MPa, (c) 90 MPa, (d) 225 MPa and (e) 450 MPa.

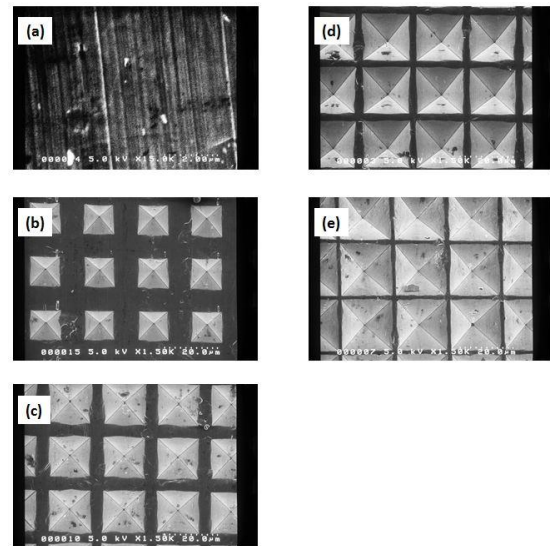


FIGURE 5. SEM images of surfaces of Al samples after nanoimprinting. Pressures between 20- μm -pitch Ni mold and Al samples: (a) 15 MPa (b) 45 MPa, (c) 90 MPa, (d) 225 MPa and (e) 450 MPa.

Figure 6 shows that the replication degree at 225 MPa is higher than 0.9. The replication degree at the pressures considered was almost the same, regardless of the pitch of the Ni mold.

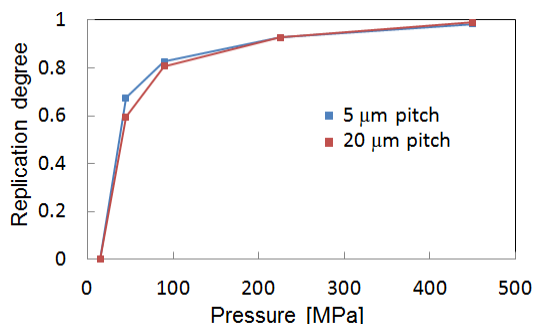


FIGURE 6. Replication degree as a function of applied pressure

Droplet Evaporation

Figure 7 shows the evaporation time (from placing a droplet on the surface of an Al sample to the disappearance of the droplet). We used nanoimprinted samples with 5- and 20- μm -pitch Ni molds and a non-nanoimprinted sample. Between 100 and 110 $^{\circ}\text{C}$, the evaporation time obtained using the nanoimprinted samples is longer than that obtained using the non-nanoimprinted sample.

The nanoimprinted samples had many more bubbles than the non-nanoimprinted sample between 100 and 110 $^{\circ}\text{C}$; however, few bubbles rose up from the samples. As shown in Figure 8, more bubbles remained on the surface of the nanoimprinted samples, and a heat insulation layer was generated. Therefore, as shown in Figure 9, the contact area decreased and a lower thermal energy was applied to the droplet than to the non-nanoimprinted sample. The droplet remained on the surface for about 30 s at 105 $^{\circ}\text{C}$ with the nanoimprinted samples. However, as the temperature became higher, bubbles flew from the sample nanoimprinted with a 20- μm -pitch Ni mold was shorter than that of the evaporation time of the sample nanoimprinted with a 5- μm -pitch Ni mold was also shorter than that of the non-nanoimprinted sample at temperatures higher than 130 $^{\circ}\text{C}$.

4. CONCLUSION

We proposed a method of fabricating nanostructures on a metal surface by nanoimprinting. We successfully demonstrated the nanoimprinting of 5 and 20 μm pyramid array patterns on aluminum. The quality of aluminum samples nanoimprinted with 5- and 20- μm -pitch Ni pyramid arrays is higher than that of a non-nanoimprinted sample at temperatures higher than 115 $^{\circ}\text{C}$ and 130 $^{\circ}\text{C}$.

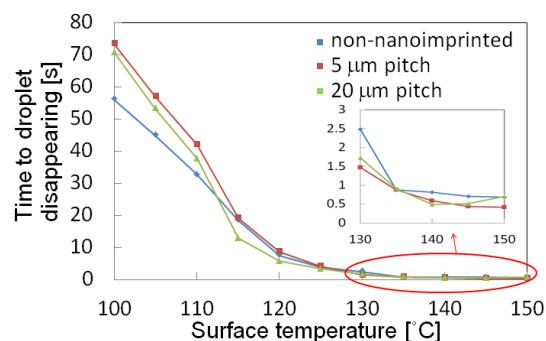


FIGURE 7. Time droplet evaporation as a function of surface temperature.

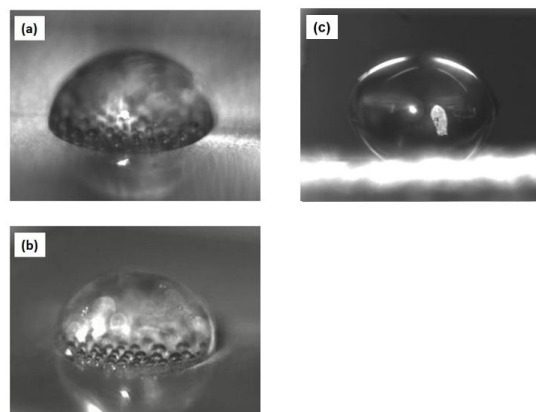


FIGURE 8. Images of droplets on surfaces of Al samples nanoimprinted with (a) 5- and (b) 20- μm -pitch Ni molds and (c) non-nanoimprinted sample at 105 $^{\circ}\text{C}$; 4 s after dropping.

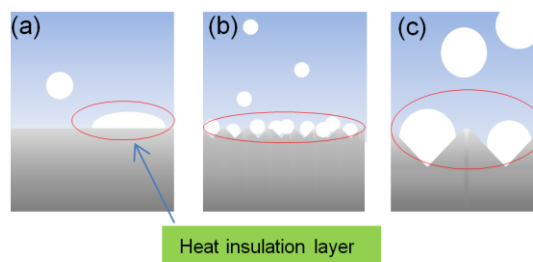


FIGURE 9. Heat insulation layer formed by bubbles remaining on surfaces of samples: (a) non-nanoimprinted sample, (b) sample nanoimprinted with 5- μm -pitch Ni mold and (c) sample nanoimprinted with 20- μm -pitch Ni mold.

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REMAINING USEFUL TOOL LIFE PREDICTIONS USING BAYESIAN INFERENCE

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INTRODUCTION

Machining science has seen tremendous gains in process modeling. Predictive model capture the physical laws governing the process. However, the model predictions are uncertain due to the inherent uncertainties in the inputs, underlying assumptions in developing the model, which may include approximations, and/or lack of information. The global marketplace has placed a premium on reducing production time and cost without sacrificing quality so model accuracy is critical. In this work, the limitation to machining productivity imposed by tool wear is addressed using Bayesian inference techniques. A new approach that establishes an estimate of the remaining useful life (*RUL*) for a selected tool based on flank wear width (*FWW*) measurements is described. Although *FWW* measurements may have limited scope in an industrial setting, the proposed method can be applied to any metric used to detect tool wear, such as spindle power or acoustic emissions during machining.

Taylor first defined an empirical relationship between tool life and cutting speed using a power law [1]:

$$VT^n = C \quad (1)$$

where V is the cutting speed in m/min, T is the tool life in minutes, and n and C are constants which depend on the tool-workpiece combination. The constant C is defined as the cutting speed required to obtain a tool life of 1 minute. Tool life is typically defined as the time required to reach a predetermined *FWW*, although other wear features (such as crater depth) may also be applied depending on the nature of the tool wear.

BAYESIAN INFERENCE

Bayesian inference models form a normative and rational method for updating beliefs when new information is available. A Bayesian model treats an uncertain variable as a random variable using a probability distribution. Let the prior distribution about an uncertain event, A , be $P(A)$, the likelihood of obtaining an experimental result B given that event A occurred be $P(B|A)$, and the probability of receiving experimental result B (without knowing A has occurred) be $P(B)$. Bayes' rule is used to determine the posterior belief about event A after observing the experiment results, $P(A|B)$, as shown in Eq. 2.

$$P(A|B) = \frac{P(A)P(B|A)}{P(B)} \quad (2)$$

The product of the prior and likelihood function is used to calculate the posterior distribution. For multiple measurements, the posterior distribution after the first measurement, or update, becomes the prior for the second, and so on. For this study, the Taylor tool life constants, n and C , are treated as uncertain, which leads to uncertainty in the predicted tool life.

BAYESIAN INFERENCE USING THE RANDOM WALK METHOD

Bayesian inference provides a rigorous mathematical framework for belief updating about an unknown variable when new information becomes available. In this study, the *RUL* of the tool was estimated using Bayesian inference and *FWW* measurements. The tool life was defined as the time required for the tool to reach a maximum *FWW* of 0.3 mm. Tests were performed using an uncoated carbide (inserted) tool to mill AISI 1018 steel. The *FWW* was assumed to increase linearly with time until the end of tool life. In the Taylor tool life model (Eq. 1), there is uncertainty in the values of the

exponent, n , and the constant, C . Subsequently, there is uncertainty in the tool life, T . The *FWW* growth curve can be predicted by generating N sample *FWW* growth curves, or sample paths, each potentially representing the true *FWW* growth curve with an equal prior probability of $1/N$. The sample paths generated in this way were used as the prior for Bayesian inference. The prior probability of the sample paths was updated by applying Bayes' rule to experimental *FWW* measurements. For each sample path, Bayes' rule can be written as the following product:

$$P(\text{path} = \text{true FWW growth curve} | \text{test result}) \propto P(\text{test result} | \text{path} = \text{true FWW growth curve}) \times P(\text{path} = \text{true FWW growth curve})$$

where $P(\text{path} = \text{true FWW growth curve})$ is the prior probability that a given path is the true *FWW* growth curve. As noted, the probability is assumed to be $1/N$ before any *FWW* measurement is completed since each path is considered equally likely to be the true *FWW* growth curve. Also, $P(\text{test result} | \text{path} = \text{true FWW growth curve})$ is referred to as the likelihood and $P(\text{path} = \text{true FWW growth curve} | \text{test result})$ is the posterior probability of the sample *FWW* growth curve given the test result. The product is divided by the normalizing constant $P(\text{test result})$. The test result denotes a *FWW* measurement.

Prior

The first step in the Bayesian inference approach is to select a prior. The prior distribution of n and C for the uncoated insert was based on previous experimental results [2]. In a separate study conducted by the authors, the means of the constants n and C for the given tool-work piece combination were found to be 0.33 and 600 m/min. Based on these values, the priors for n and C were selected to be uniform distributions with minimum values of 0.3 and 400, respectively, and maximum values of 0.35 and 700, respectively, independent of each other. This is expressed using: $n = U(0.3, 0.35)$ and $C = U(400, 700)$, where U denotes a uniform distribution and the values in the parentheses identify the minimum and maximum values, respectively. The random sample *FWW* growth curves were generated as follows. First, random samples were drawn from the prior joint probability density function (pdf) of n and C and the Taylor tool life curve was calculated for each

$\{n, C\}$ pair; this exercise was repeated 1×10^4 times. As noted, the *FWW* was assumed to increase linearly with time until the end of tool life. Second, the linear *FWW* growth curve was generated for all the tool life values calculated from the first step at the selected experimental spindle speed. These growth profiles were the prior random sample paths used for Bayesian inference. Figure 1 shows the prior histogram of tool life at 5000 rpm. Figure 2 shows the prior cumulative distribution function (cdf) of *FWW* as a function of machining time at 5000 rpm. Figure 3 shows the prior probability of *FWW* being less than 0.3 mm (defined as the *FWW* limit) as a function of machining time. From Figure 3, the 95% *RUL* of the tool was 2.7 minutes. This implies that there is a 5% probability that the *FWW* will exceed 0.3 mm at 2.7 minutes.

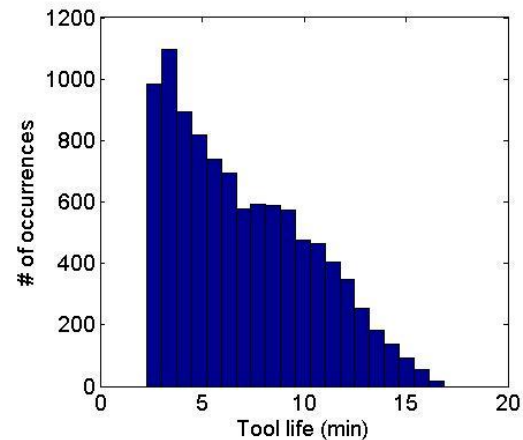


FIGURE 1. Prior histogram of tool life at 5000 rpm.

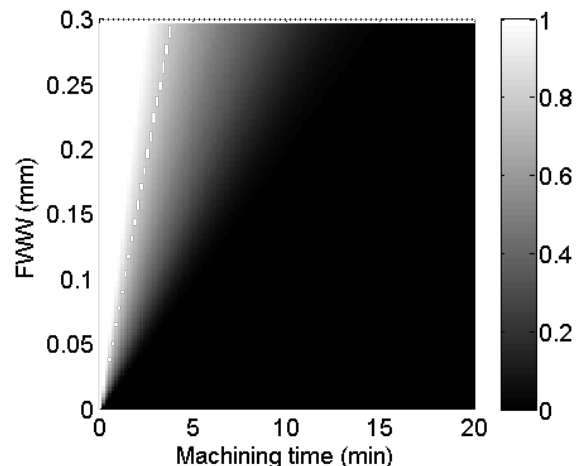


FIGURE 2. Prior cdf of *FWW* as a function of machining time at 5000 rpm.

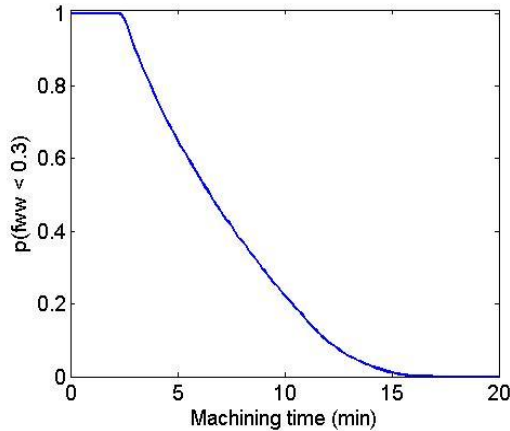


FIGURE 3. Prior probability of FWW being less than 0.3 mm as a function of machining time.

Likelihood

The probability of the sample FWW growth curves was updated using FWW measurements and Bayes' rule. The likelihood function incorporates the uncertainty in tool wear and the assumed linear FWW growth model. A non-normalized Gaussian distribution was used as the likelihood in his study shown in Eq. 3:

$$l = e^{-\frac{(fww - fww_m)^2}{k}} \quad (3)$$

where l is the likelihood value, fww_m is the measured FWW , fww is the FWW value for a sample curve at the experimental spindle speed, and k depends on the tool wear uncertainty. Because the likelihood function is expressed as a non-normalized normal distribution, $k = 2\sigma^2$, where σ is the standard deviation of fww ; it represents the uncertainty in tool wear and the linear FWW growth model. The likelihood function describes how likely is the given the FWW measurement result, at a particular machining time, given that the sample FWW growth curve is the correct curve. Figure 4 shows the likelihood function for a measured FWW of 0.1 mm after 5 minutes. If the FWW growth curve value is near the measurement result, then the likelihood value is high. Otherwise, it is low. The likelihood function can be interpreted as assigning weights from 0 to 1 to the sample curves; 0 implying not likely at all and 1 implying most likely. As shown in Figure 4, increased uncertainty (higher σ) widens the likelihood function so that comparatively higher weights are assigned to sample curves far from the experimental result. Subsequently, larger

uncertainty yields a more conservative estimate of tool life.

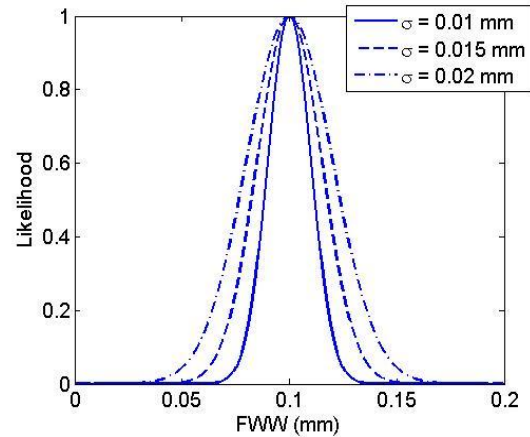


FIGURE 4. Likelihood values for various uncertainty levels for a measured FWW of 0.1 mm after 5 minutes of machining.

Bayesian updating

According to the Bayes' rule, the posterior probability is the product of the prior and the likelihood and normalized so that the sum is equal to unity. The prior probability for each path is 1×10^{-4} and the likelihood value is determined using Eq. 3; the value of σ was taken as 0.01 mm. The updated probabilities of the sample paths, calculated as the product of the prior and likelihood and normalized such that the sum is equal to unity, were used to determine the posterior FWW cdf. Figure 5 shows the updated cdf of FWW given a measurement of 0.1 mm FWW at 5 minutes. Figure 6 shows the posterior probability of FWW being less than 0.3 minutes. The 95% RUL was 7.4 minutes; note that the measurement was taken after 5 minutes.

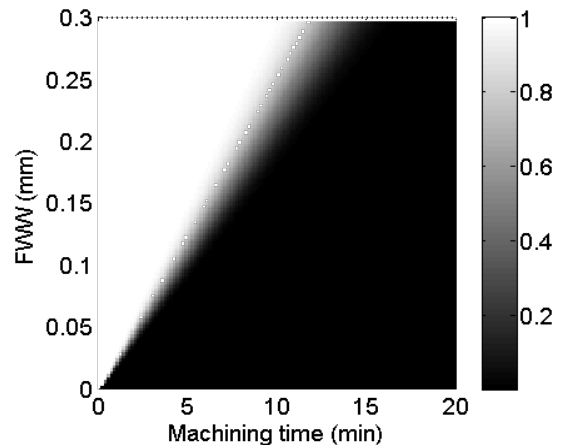


FIGURE 5. Posterior cdf of FWW as a function of machining time at 5000 rpm.

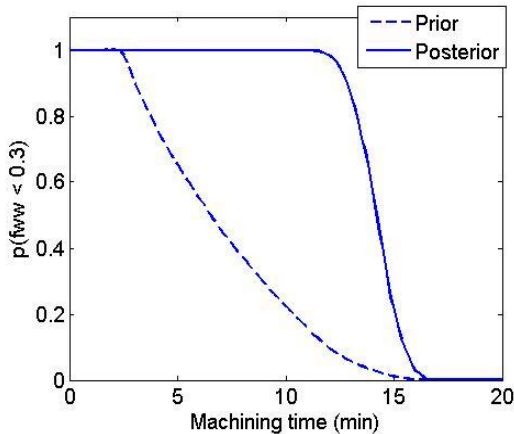


FIGURE 6. Posterior and prior probability of FWW being less than 0.3 mm as a function of machining time.

RUL PREDICTIONS

Tests were performed using uncoated carbide insert and 1018 steel workpiece material. The FWW was measured at regular intervals using a portable digital microscope. Figure 7 shows the growth of FWW with machining time. The tool life was found to be 11.9 minutes. The measured FWW values were then used to update the prior probabilities of the sample FWW growth curves using the procedure described previously. The value of σ was assumed to be 0.01 mm. Note that the updated probabilities of sample paths were used to calculate the posterior cdf of FWW. From the updated probabilities of the sample FWW growth curves, the 95% RUL of the tool before the FWW reaches 0.3 mm was determined. Each FWW measurement updates the RUL estimates. Figure 8 shows the 95% RUL of the tool after each measurement and the true remaining life calculated from the observed tool life value (11.9 minutes). The RUL estimates approach the true remaining life from the conservative side.

CONCLUSIONS

The application of Bayesian inference to RUL predictions of the tool was demonstrated using a random walk approach, where the prior probability of FWW was generated using sample FWW growth curves that represented the true FWW growth curve with some probability. This probability was updated using Bayesian inference. Although a linear FWW growth model was assumed in this study, a higher order model may also be assumed to describe the three stages of tool wear [3]. The method can be extended to include sensor data such as power

or acoustic emission. In addition, uncertainty regarding the threshold value of the sensor, such as percent increase from the nominal, can also be incorporated.

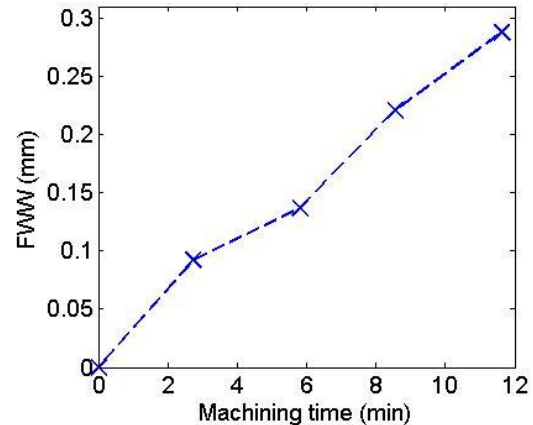


FIGURE 7. Experimental FWW growth with machining time.

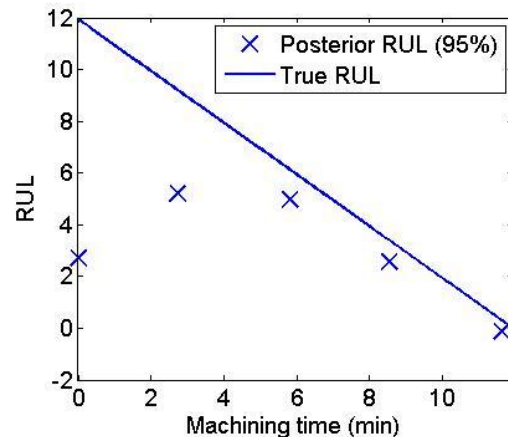


FIGURE 8. 95% RUL predictions.

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Mechanisms of CMP Pressure Profile Generation

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INTRODUCTION

The purpose of this work is to generate an analytical model of the physical mechanisms that govern the wafer pressure profile in chemical mechanical polishing (CMP). This understanding makes it possible to reliably and simply reduce the within-wafer non-uniformity (WIWNU) to <3% over the whole wafer. The improved polishing uniformity will allow for finer control of film thicknesses as well as a marked increase in wafer area utilization, translating into cost reductions for integrated circuits which require CMP.

MOTIVATION

It is presently infeasible to utilize the far edges of silicon wafers as CMP produces significant etching variations within the outer $\approx 5\text{mm}$ of the wafer [1,2], which can account for 5% of the wafer area. This reduces the number of usable integrated circuits (ICs) per wafer, increasing costs. The material removal rate on the wafer is a complex function of the chemical slurry, wafer relative velocity and multiple pressures applied to the CMP head and wafer, making it difficult to control variation over the wafer [3] or determine the source of variation over the wafer with the present theoretical understanding.

BACKGROUND

Previous work on CMP contact mechanics and etch rates has mainly used closed form expressions [2] and 2D axi-symmetric FEA models [1,2,4,5] to predict contact pressure. Preston's equation links the wafer-pad contact pressure to the etch rate, allowing for an estimation of this rate [3]. A spike in pressure is normally observed at the outer few mm of the wafer due to geometric stress concentrations [1,2,4] which increases the WIWNU of the CMP process.

A retaining ring is used to restrain the slip of the wafer during CMP. This ring is pressed into the pad. The effect of this retaining ring on the wafer edge pressure profile is significant [1,5], however there is little discussion of the

mechanism governing this sensitivity and how it can be utilized in a manufacturing environment.

MODEL

A 2D axi-symmetric model was generated using Solidworks Simulation to study CMP contact mechanics. Sliding rather than bonded contact was used between the 300mm wafer/pad, wafer/membrane and retaining ring/pad. CMP head designs [6] often use a metal ring to pass load down the wafer surface, as seen in Fig. 1 and 4.

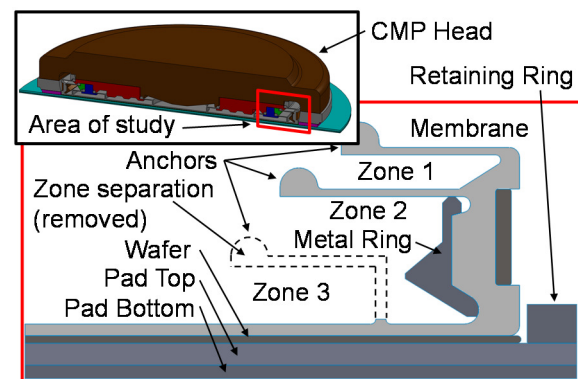


FIGURE 1. Side view of the 2D axi-symmetric model used for this analysis. The CMP device structure is included in this analysis to study the propagation of loads into the wafer. A multi-zone membrane is modeled above the outer 5cm of the wafer.

LOAD TRANSFER MECHANISMS

The pressure transfer mechanisms through the wafer are usually treated as a direct conversion from pressure above the wafer to contact pressure below. This research suggests that a more effective model of this transfer includes an intervening displacement domain, as shown in Fig 2.

The membrane and retaining ring pressures are scaled through their own mechanisms that transform these loads into the displacement domain. The wafer motions corresponding to this domain are shown schematically in Fig. 3, including a wafer translation into the pad z_{wn} , a

wafer cantilevering of angle θ_w , and the pad edge displacement Δz_e , which measures the shear stress in the pad as the difference between the wafer edge depth z_{we} , and the pad equilibrium depth at the edge z_{pe} .

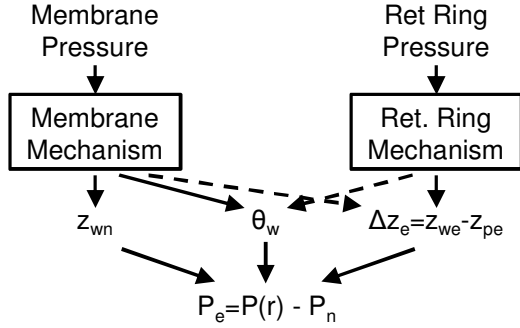


FIGURE 2. Proposed contact pressure generation mechanism with intervening displacement regime.

The membrane plays a minor role on the edge pressure, and the retaining ring plays a correspondingly minor role on the wafer cantilevering. Each of these displacements generates a unique pressure profile as shown in Fig 3.

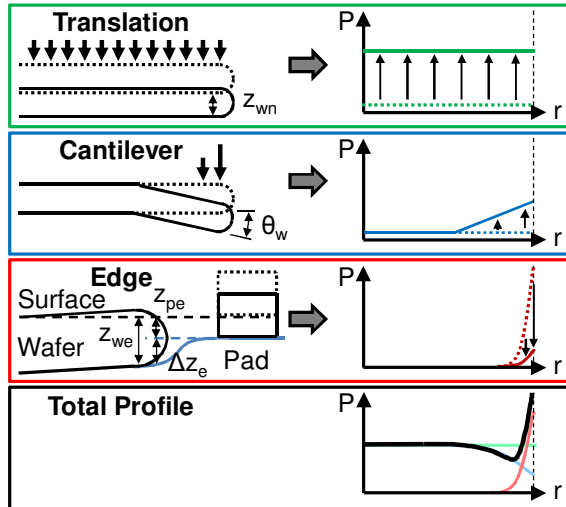


FIGURE 3. Dominant wafer kinematics and governing variables with the associated characteristic pressure profile. The sum profile is shown at the bottom.

WAFER TRANSLATION

The wafer translation describes bulk motion of the wafer into the pad. This is governed by the nominal pressure P_n , applied over the whole wafer, and the pad stiffness k_{pz} , which is 0.147

psi/ μ m for a IC1000 pad. E and t are the Young's modulus and thickness of the top and bottom parts of the pad, while ν is the average Poisson ratio.

$$z_{wn} = P_n / k_{pz} \quad k_{pz} = \left(\frac{E_t}{t_t} \parallel \frac{E_b}{t_b} \right) \frac{1-\nu}{1-\nu-2\nu^2} \quad (1)$$

WAFER CANTILEVERING

The slope in the outer wafer profile is regulated by the membrane zone 1 pressure. This is not applied directly to the surface, so is often subject to significant tuning during optimization. The zone 1 pressure passes through a rigid internal ring down to the surface of the wafer (Fig. 4), appearing as a spike in Fig. 5.

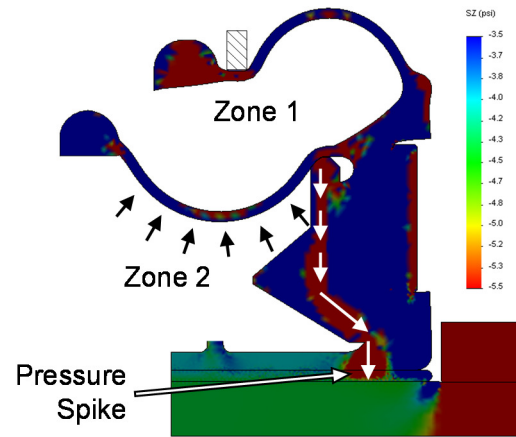


FIGURE 4. Pressure transfer within the membrane mechanism, with the zone 1 flow highlighted.

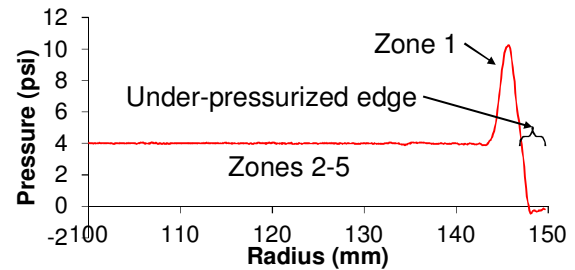


FIGURE 5. Membrane-wafer pressure profile showing both the zone 1 spike and the under-pressurized edge.

This pressure (in excess of nominal) acts as a torque to rotate the outer wafer around a effective joint at $r_j = 130.3$ mm radius, with rotary stiffness $k_\theta = 2899$ Nm/rad, shown in Fig. 6. This effect is generated through the innate stiffness of the wafer relative to the pad.

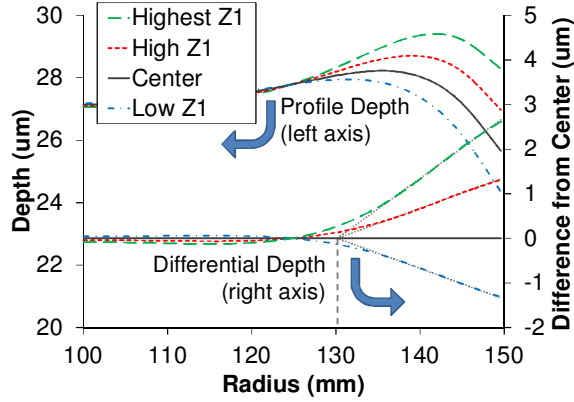


FIGURE 6. Wafer displacement (top with left axis) due to varying the zone 1 pressure. The deviation from centered is also plotted (bottom with right axis) to clearly show the effective joint at 130.3mm. Note the centered profile shows edge curvature due to the under-pressurized edge.

The torque on the pseudo-joint, τ , can be calculated as Eq. 2, where $P_m(r)$ is the membrane-wafer pressure as seen in Fig. 5, and $P_e(r)$ is the edge pressure defined in the next section. This is a second order correction, as it will generally be near zero for a well-tuned system. τ^* is defined as τ without the $*$ term.

$$\tau = 2\pi \int_{r_j}^{r_o} \left(P_m(r) - P_n - \overbrace{P_e^*}^* (r) \right) (r - r_j) r dr \quad (2)$$

EDGE

The edge pressure spike is driven by shear stress in the pad, compressing the pad beyond the edge of the wafer. This is captured by the edge stiffness k_{pze} below. The spike amplitude P_e scales with the difference between z_{we} and z_{pe} . The pad depth is driven by the retaining ring pressure P_r , acting across the wafer-retaining ring gap g .

$$P_e = k_{pze} \left(\frac{P_n}{k_{pz}} + \frac{\tau^*}{k_\theta} \left(r_o - r_j \right) - \frac{P_r}{k_{pz} + k_{pze}} e^{-\frac{g}{\tau_p}} \right) \quad (3)$$

This satisfies the sanity check of surface continuity (z_{we} and z_{pe}) requiring pressure continuity ($P_e \rightarrow 0$). The cantilevering is a second order term that will be near zero for a well-tuned system. P_e^* is defined as P_e without the $*$ term. This expression has been tuned against simulation, shown in Fig. 7.

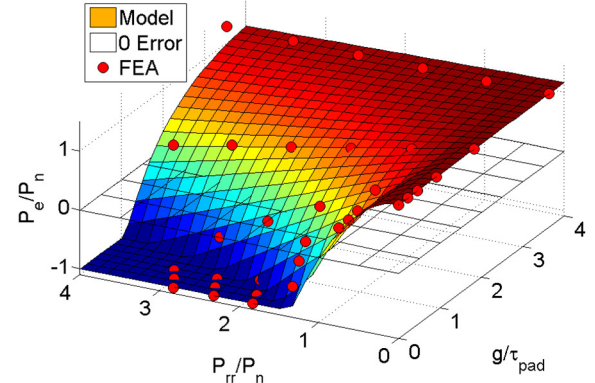


FIGURE 7. Comparison between edge pressure theory and FEA simulation- shown as points.

The theory in Eq. 3 was compared against FEA, allowing the definition of $c_{ke} = k_{pze}/k_{pz} = 1.17$, $c_{kr} = k_{pze}/k_{pz} = -0.45$, and the pad decay constant, τ_p , as 1100μm. Real wafer etching data suggests τ_p is actually closer to 425μm.

The edge pressure profile $P_e(r)$ is assumed to be an exponential decay due to the physics of shear transfer. A double exponential term was utilized in Eq. 4 to capture the spike and overshoot observed in the differential pressure profiles shown in Fig. 8. These define the ratio of the overshoot amplitude to spike amplitude, c_e , as 0.2, the edge profile decay constant, τ_{pe} , as 500μm, and the ratio of overshoot decay to spike decay constants, c_r , as 4.

$$P_e(r) = \frac{P_e}{1 - c_e} \left(e^{-\frac{r_o - r}{\tau_{pe}}} - c_e e^{-\frac{r_o - r}{c_r \tau_{pe}}} \right) \quad (4)$$

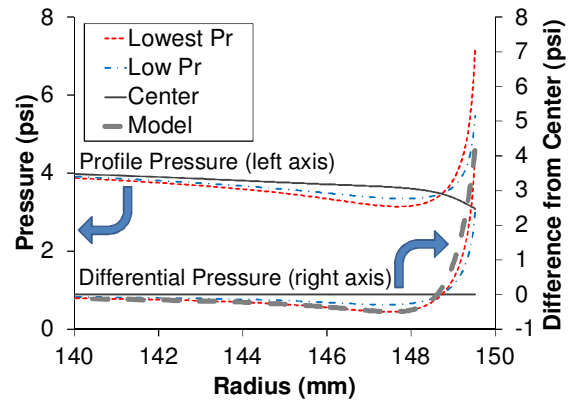


FIGURE 8. Wafer pressure (top and left) due to varying the retaining ring pressure. The deviation from centered is also plotted (bottom and right) to show the edge pressure profile with the exponentially decaying spike and overshoot. The profile function is fit to the larger spike.

$$P(r) = k_{pz} \left(\overbrace{\frac{P_n}{k_{pz}} + \frac{\tau}{k_\theta} \langle r - r_j \rangle^1 + \dots}^{z_w(r)} \right) + \frac{c_{ke}}{1 - c_e} \left[P_n + \frac{k_{pz} \tau^*}{k_\theta} (r_o - r_j) - \frac{P_r e^{-\frac{g}{\tau_p}}}{1 + c_{kr}} \right] \left(e^{-\frac{r_o - r}{\tau_{pe}}} - c_e e^{-\frac{r_o - r}{c_\tau \tau_{pe}}} + \dots \right) \quad (5)$$

FULL PROFILE

The total pressure profile, $P(r)$, is the superposition of the pressure profiles generated for each of the three displacement sources, as shown in Eq. 5. A first order singularity function is used to approximate the wafer cantilever effect. More complex functions could be used to increase the model accuracy at predicting the wafer displacement profile, $z_w(r)$, around the effective joint. Further increases in accuracy could be found by increasing the complexity of the edge pressure profile expression.

RESULTS

Our theory shows weak coupling between the wafer cantilevering and edge spike effects. Additionally the error terms- cantilevering and edge pressure- can be driven to zero, leaving nearly only nominal pressure over the wafer. This indicates a simple method for practical tuning of wafer profiles, whether in FEA or an actual manufacturing setting.

- 1) Set the nominal pressure P_n to the desired etch rate, and set zone 1 to generate 0 rotation based on the τ^* expression in Eq. 2.
- 2) Use the retaining ring, P_r , to remove the edge spike, moving over the surface in Fig. 7 until $P_e/P_n \approx 0$ is reached.
- 3) Use zone 1 to cantilever the wafer until the remaining second order error effects (seen in Fig 6) are minimized.

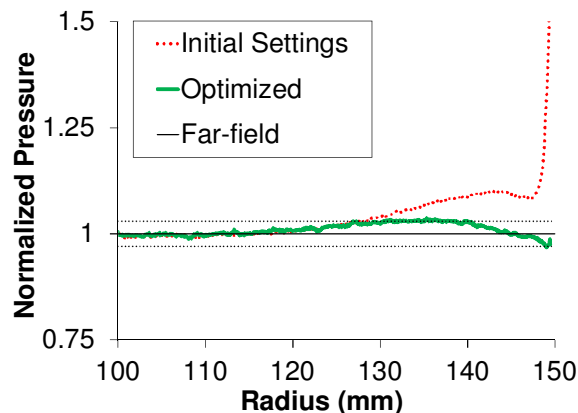


FIGURE 9. Reduction in edge variation through use of pressure transfer mechanism models to set membrane and retaining ring pressures.

This process was done in FEA for a wafer with common settings, shown in Fig 9. The error was brought down to within 3% (dotted lines on chart) over the whole wafer. The remaining error in the profile is due to the under-pressurized edge shown in Fig. 5.

CONCLUSION

In this work we develop an analytical model of the contact mechanics governing pressure profile generation in CMP. This understanding makes it possible to i) improve CMP head designs, and ii) develop a simple and reliable process for improving pressure uniformity, increasing the usable area of the wafer. This is demonstrated by using the process to reduce WIWNU to <3% over the entire wafer. This increases the usable wafer area to nearly 100%.

ACKNOWLEDGEMENTS

We thank Samsung for their support.

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AN ANALYSIS OF POLISHING FORCES IN MAGNETIC FIELD ASSISTED FINISHING

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INTRODUCTION

In magnetic field assisted finishing (MAF) a magnetic field is used to maneuver a flexible “magnetic brush” (composed of ferromagnetic particles and abrasives captured in a magnetic field) over the surface to be polished. The relative motion between the brush and the surface can be obtained either by rotating the brush, moving the sample, or both. The brush can consist of either: 1) sintered particles, where the ferromagnetic and abrasive particles (e.g., SiC, Al₂O₃, CBN, or diamond) are sintered together to form a ferromagnetic conglomerate; or 2) separate abrasive and ferromagnetic particles. For the latter case, the abrasives are held between and within the magnetic chains (brush) which are formed. A lubricant can also be used to aid in holding the abrasive particles within the flexible brush. Figure 1 provides a schematic of the material removal process.

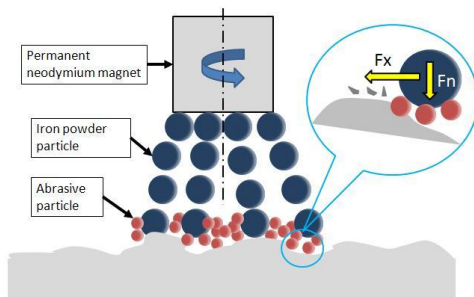


FIGURE 1. MAF process.

In this study, the polishing forces during MAF were examined. The measured forces were divided into forces due to polishing effects and forces due to magnetic effects. A method to identify and isolate the different elements of the force measurement was established. The effect of varying the iron particle size and the working gap on polishing forces and surface roughness was examined. A coherence scanning interferometer (CSI) was used to measure surface roughness. A surface roughness of 2-3 nm Sa was achieved under certain conditions.

EXPERIMENTAL SETUP

The experimental setup consisted of mounting a permanent neodymium magnet in a fixture held in the spindle of a CNC milling machine. The sample was mounted on a ferromagnetic mount which completed the magnetic circuit and strengthened the magnetic field in the gap between the magnet and the sample. This increased the forces on the iron particles within the slurry. The sample was glued to the sample mount using an epoxy resin. The sample-sample mount combination was fixed on a 3D force sensor (Kistler 9256 C1 dynamometer). Each polishing trial consisted of approaching the sample in the normal direction to capture all the magnetic effects. The magnet was then rotated and translated for a fixed period before retracting from the sample. Figure 2 shows a schematic of the experimental setup.

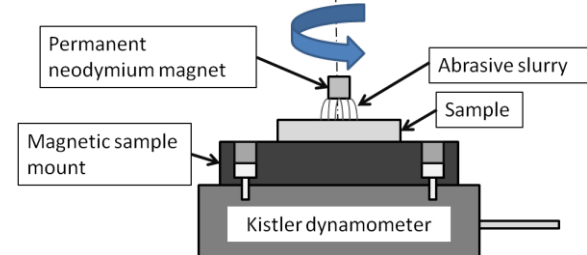


FIGURE 2. Schematic of the experimental setup.

Polishing Forces

The MAF forces can be divided into two categories: 1) magnetic effects; and 2) interactions between the brush and sample. When studying the forces for different polishing conditions, it is important to accurately separate the two categories. The magnetic forces exist mainly due to magnetic interactions between the permanent magnet, the ferromagnetic sample mount, the ferromagnetic slurry and any magnetism the sample may exhibit. Each of

these effects was individually identified using a series of tests. Once the cumulative magnetic effects were evaluated, a reference was computed with respect to which the polishing forces were measured. The individual tests are listed below

A) No Sample, No Slurry

The polishing cycle was completed with no sample or slurry. This enabled the pull of the magnet on the sample mount in the surface normal direction (n) to be isolated.

B) No Sample, With Slurry

The polishing cycle was completed without the sample, but with the slurry. There was still no polishing because the slurry did not make contact with the sample mount. This enabled the effect of the slurry on the normal force to be identified.

C) With Sample, No Slurry

The polishing cycle was completed with the sample, but without the slurry. There was therefore no polishing. This test enabled the extra pull due to the sample to be determined. The individual magnetic effects were then identified using the following relationships.

i) Effect of sample mount:

$$F_{sm} = -F_{n(\text{no sample, no slurry})}$$

ii) Effect of slurry:

$$F_{sl} = F_{n(\text{no sample, slurry})} - F_{n(\text{no sample, no slurry})}$$

iii) Effect of sample:

$$F_{sa} = F_{n(\text{sample, no slurry})} - F_{n(\text{no sample, no slurry})}$$

The reference was then computed using the following relationship.

$$\text{Reference} = -F_{sm} - F_{sl} - F_{sa}$$

Figure 3 shows a plot of the normal forces for the different tests used to separate the polishing and magnetic forces. The different forces are identified in figure 4. The magnetic forces exist mainly in the normal direction along the axis of the magnet. The symmetry that exists in the two orthogonal directions nullifies the magnetic effects and only polishing forces are observed. Tests were also conducted to evaluate the effect of translational velocity on lateral polishing forces. No significant effects were observed.

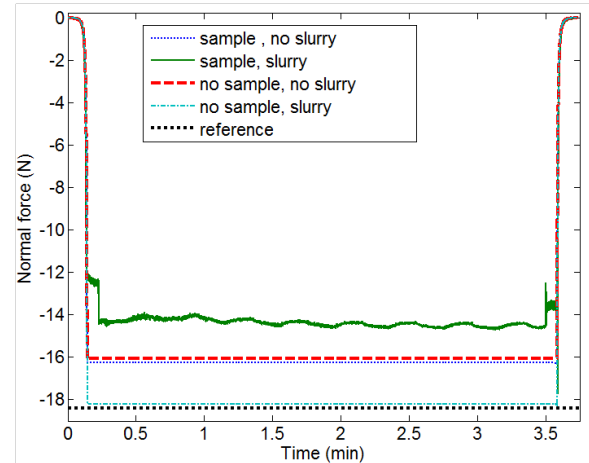


FIGURE 3. Normal polishing forces.

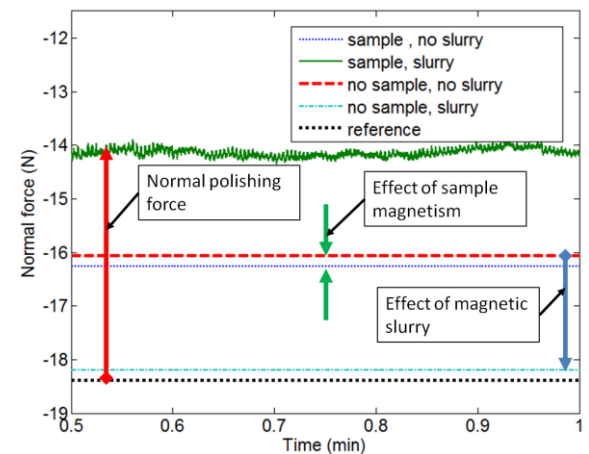


FIGURE 4. Normal force effects.

Surface Roughness

The surface roughness was measured using the CSI at nine different locations on the sample after each polishing trial. Filtering was completed in accordance with ISO 4288. In order to correctly compare the performance of different polishing conditions it was necessary to establish a common starting roughness on all samples. The samples were initially polished using a nylon mesh pad to a consistent surface roughness of 70-90 nm Sa.

RESULTS

Polishing Forces

The normal force magnitude decreased slightly as the iron particle size increased. It is proposed that the smaller iron particles enabled the formation of a more even brush which caused a larger number of particles to be in contact with the sample. Therefore, although the smaller iron particles may experience smaller magnetic field forces individually, on the whole they apply more

attractive force between the magnet and sample resulting in a higher normal force. This effect was also seen in the lateral force measurements. These results are displayed in figure 5.

The normal polishing force exhibited a strong dependence on the working gap. This is because the flux density decreases as the working gap increases. The lateral forces were also inversely related to the working gap. The results are shown in figure 6.

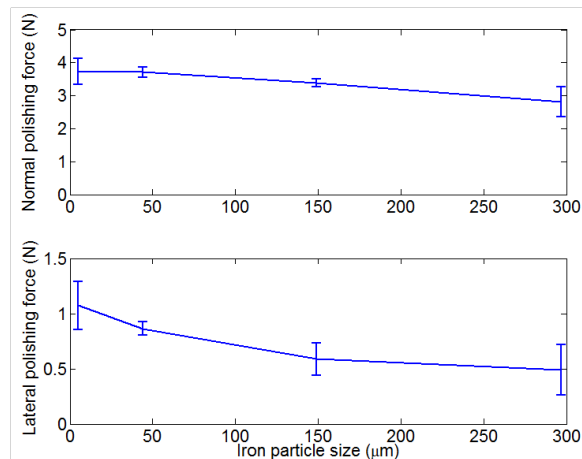


FIGURE 5. Polishing force measurements for varying iron particle size.

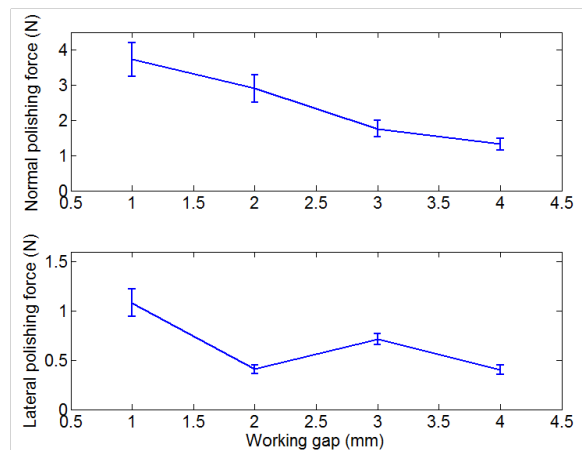


FIGURE 6. Polishing force measurements for varying working gap.

Surface Roughness

There is no material removal when the iron particle size (IPS) is smaller than the top width of the valley. As the IPS increases, the particles are prevented from entering the valleys and produce material removal at the peaks. It is also important to note that, although the entire brush

might produce larger normal and lateral forces, each individual particle for the smaller sized iron particles exerts little force on the surface. Thus, the localized polishing force is higher when IPS is greater. This resulted in a faster improvement in surface roughness.

A combination of two factors dictates whether there is material removal: 1) the IPS should be larger than the top width of the valleys in the surface; and 2) the IPS must be large enough to impart sufficient force on the individual abrasive particles to cause material removal. The material removal rate and surface roughness improve significantly for larger IPS (Figure 7). The working gap (WG) was found to have little effect on polishing performance as long as the amount of slurry was maintained proportional to the WG (Figure 8).

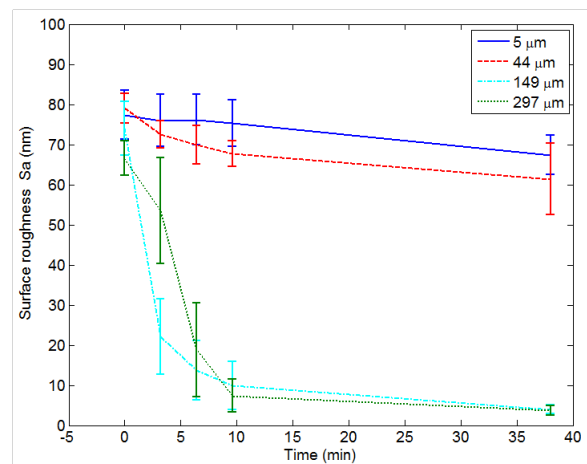


FIGURE 7. Surface roughness measurements for varying iron particle size.

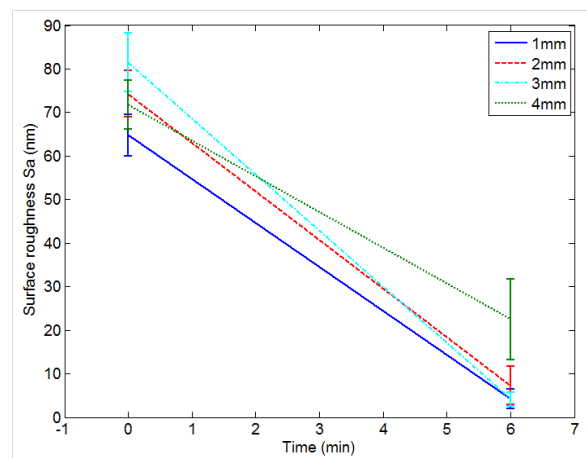


FIGURE 8. Surface roughness measurements for varying iron particle size.

Material Removal Rate (MRR)

The effect of iron particle size on the MRR for MAF was analyzed. The material removed was determined by imaging the sample surface on the CSI before and after polishing. A set of dimples were created on the sample surface using a spherical indenter. The spherical indenter ensured low slopes at the bottom of the indent which made it possible to image the bottom of the indent. These indents were imaged on the CSI to measure the depth of the indent. The bottom of the indent was set as the reference surface with respect to which the material removal rate was then determined. It was assumed that there was no material removal at the bottom of the dimple.

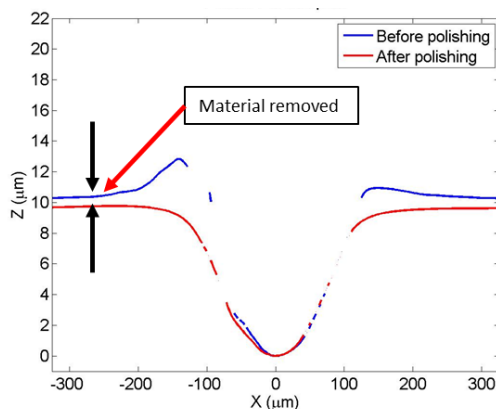


FIGURE 9. Cross-section of dimple.

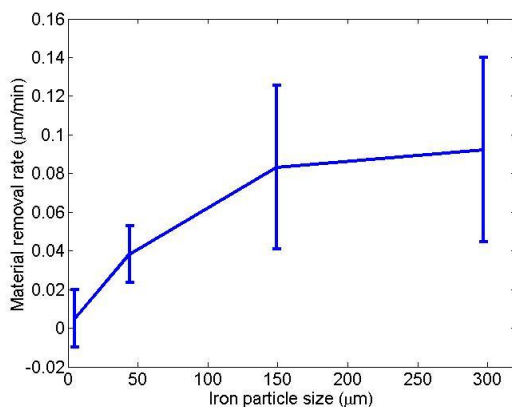


FIGURE 10. Material removal rate for varying iron particle size.

Figure 9 shows the cross-sectional views of the dimple before and after polishing along with a representation of the material removed. The MRR was measured for polishing with different iron particle sizes; see figure 10. The MRR was measured at nine different locations on the

sample. The error bars represent one standard deviation in MRR between these nine locations.

Surface Texture

The direction of lay on the polished surface depends on the path of the slurry in contact with the surface. A direct correlation between the direction between the abrasive path and surface texture was observed.

CONCLUSIONS

A number of conclusions can be drawn based on the experimental results and analysis.

1. While measuring MAF polishing forces, it is necessary to isolate the magnetic field effects from the polishing effects.
2. The normal and lateral forces were found to have an inverse relationship with both IPS and WG.
3. The rate of improvement in surface roughness with polishing time depends on the IPS. If the particles are too small, there is little material removal even after polishing for prolonged periods of time. The rate of improvement in surface roughness is not sensitive to WG.
4. The MRR increases considerably with iron particle size.
5. The translational velocity has almost no effect on the magnitude of lateral polishing force.
6. The lay of the polished surface is dictated by the path of the magnetic field and the direction of the abrasive particles.

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INTEGRATED MICROFLUIDIC DEVICE FOR FORENSIC FIBER DYE ANALYSIS

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INTRODUCTION

Due to the quantity of textile materials in the environment, there is a high probability of fiber transfer during the commission of a crime. As a result, identification of fiber samples often plays a critical role in criminal investigations. In a typical case, a sample of known origin must be compared to an evidence sample to determine if both have the same source. A wide variety of analytical techniques are available to test this hypothesis and each provides a different type of information. The fiber examiner typically begins with non-destructive microscope techniques which discriminate samples based on morphology (e.g., fiber shape, color, etc.) [1]. If additional specificity is needed to prove the hypothesis, more sensitive analytical techniques such as Liquid Chromatography-Mass Spectrometry (LC-MS) are employed but typically destroy the evidence sample [2]. Prior to the use of these analytic techniques, dye molecules must be separated from the fiber sample. This paper describes the development of a microfluidic system that automatically extracts the dye from a fiber strand less than 2 mm in length and prepares it for analysis.

The process starts with a small fiber sample placed into an extraction chamber by a forensic examiner. The analysis is automated from that step forward including:

1. Securely holding the fiber
2. Sealing the extraction chamber
3. Observing the fiber in place after insertion
4. Selecting the appropriate extraction solvent depending on fiber and dye type
5. Metering the solvent into the chamber
6. Heating and evaporating the solvent from the chamber
7. Dissolving the extracted dye with a elute solution
8. Sending the dye solution into the LC-MS system for identification
9. Presenting the results to the examiner
10. Controlling the entire system.

The introduction of this system into a crime lab will lead to less contamination and faster analysis of fiber dye samples. In a broader sense, this microfluidic technology can be applied to other chemical processes from forensic science to medical laboratory to environmental analysis. This project is an interdisciplinary effort combining expertise from the Colleges of Engineering and Textiles with input from the North Carolina Crime Laboratory.

MICROFLUIDIC DESIGN

The first prototype device for dye extraction is shown in Figure 1. The fiber is captured between two 200 μm high posts in the chamber. Solvent is introduced through channel 3 to fill the chamber and extract dye from the captured fiber. Solvent is then heated (heater under chamber) and evaporated by flowing air into channel 1 and out channel 2. An analyte solution is introduced via channel 4 and finally, the dissolved dye is pushed out through channel 5 directly into an LC-MS instrument to separate and identify the dyes.

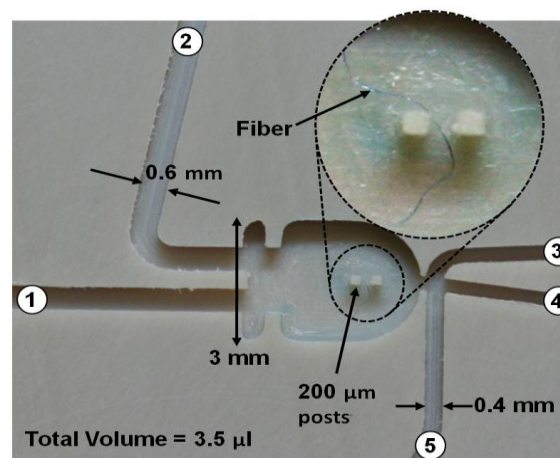


FIGURE 1. Layout of dye extraction chip machined in PEEK Polymer material. The fiber is captured between two 200 μm posts. Channels are 1) gas exhaust, 2) gas inlet, 3) solvent inlet, 4) analyte inlet, and 5) port to mass spectrometer.

The extraction system is driven by a microprocessor that controls the process. Based on the fabric and dye, a particular chemistry is selected, a timed volume of the extraction solvent is sent to the chip, the heater is turned on and the evaporation air is circulated over the fluid. Chamber shape and flow parameters are optimized using computational fluid dynamics as shown in Figure 2 to minimize evaporation time (the longest step in the extraction process) without expelling any of the liquid, and hence the dye molecules, from the exhaust port.

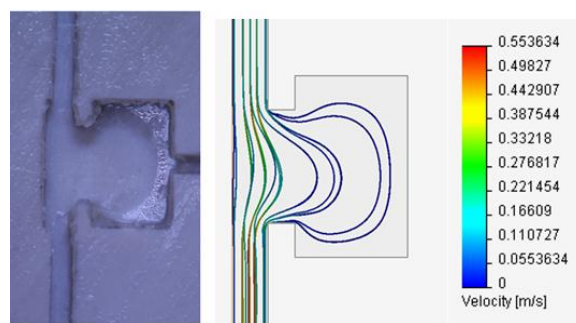


FIGURE 2. CFD analysis allowed refinement of channel and chamber designs and dimensions.

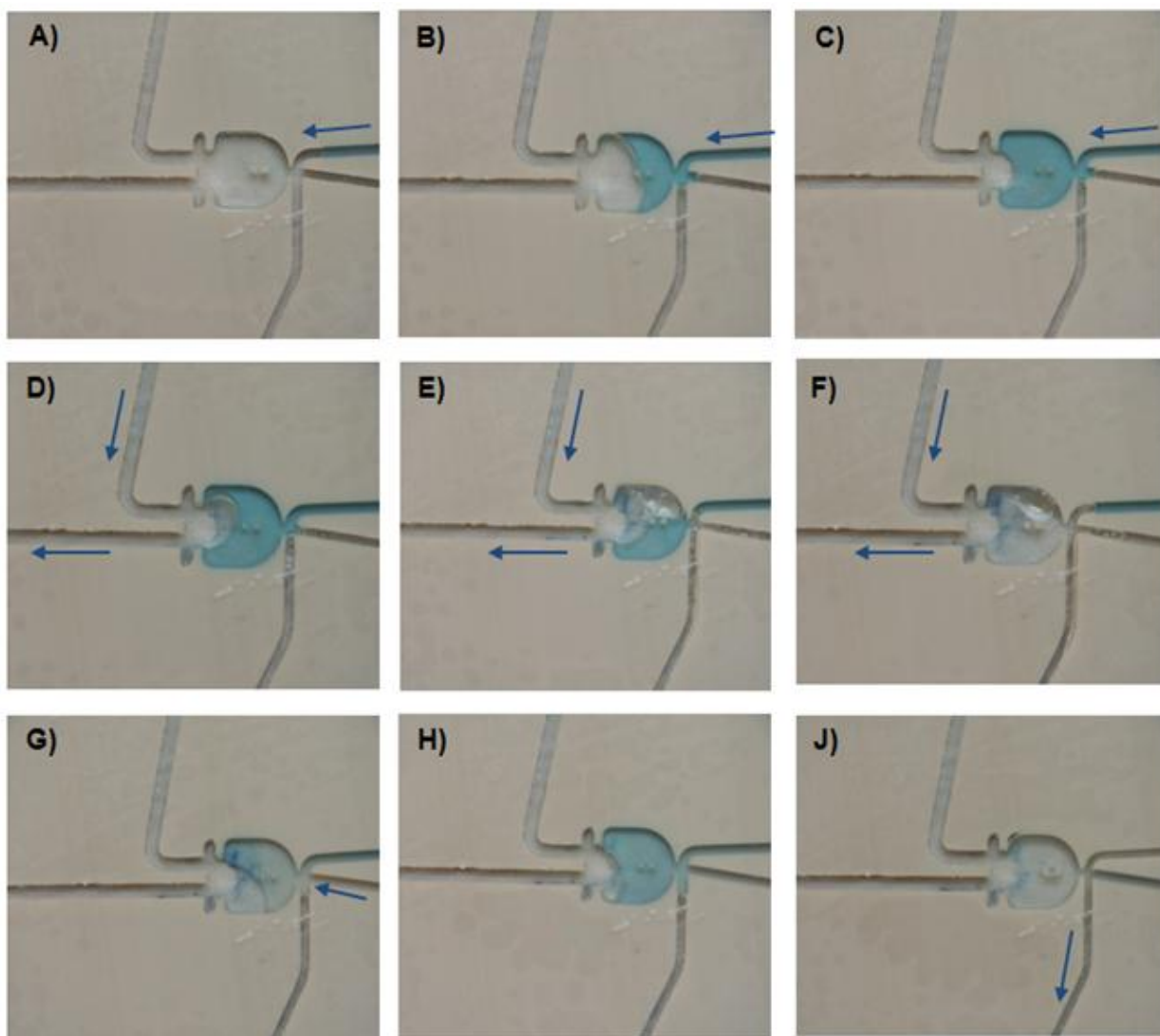


FIGURE 3. The complete extraction process. The 4:3 pyridine:water solution is injected (A-C). Gas (air) is then flowed across the chamber to evaporate the solution (D-F). ACN is then introduced (G) to reconstitute the dye (H). Finally, the ACN/dye solution is expelled to the Mass Spectrometer (J) using air pressure.

Once the extraction solvent is evaporated the dye molecules will be left on the surface of the chamber. A known volume of dye elution solution (ACN/H₂O) is then sent into the chamber where it will dissolve the dye and pass into the input channel to the LC-MS. The combination of these two instruments will show both the time when the molecules pass through the LC column and the actual mass/charge ratio that identifies the dye molecule.

Extraction tests were performed by injecting the extraction solvent (4:3 pyridine:water solution) into the chamber as shown in Figure 3(A-C). The resultant dye/pyridine solution was then heated to 100°C using the PTC heater and exposed to a flow of dry air for 10 minutes shown in Figures 3(D-F). Once the pyridine evaporated, the extracted dye was dissolved in an acetonitrile (ACN)/water solution, which was injected through another port (Figure 3(G)). The dye/ACN solution was then reconstituted and pushed through another port to the for LC/MS analysis (Figure 3(J)). The prototype device has a total fluid volume of 3.5 µl and was used to demonstrate the entire process. A 2 mm long fiber is captured between two 200 µm posts. The curl of the fiber held it in the extraction chamber between the two posts. The complete extraction process takes about 10 minutes.

Identification of a dye was demonstrated with this system using a polyester fiber. The dye was a standard Disperse Blue 60 dye. The extraction process used a pyridine volume of 4 µl (1 µl is a cube with 1 mm on each side) and a total time from fabric insertion to extracted dye ready for the LC-MS of less than 10 minutes. The Blue 60 dye has two different molecules which have a mass of 380.12 Da and 424.15 Da. These molecules were identified unambiguously using first LC separation, then UV-VIS spectrometry and finally the MS.

SYSTEM DESIGN AND CONTROL

A schematic diagram of the proposed system layout is shown in Figure 4. A series of extraction solvents contained in pressurized vessels will be available in the inlet area and will be selected based on the fabric/dye combination. A series of valves control the flow of solvents and air to and from the microfluidic chip. Also controlled are the PTC Thermistor used to heat the chip to 100°C for accelerated evaporation and a air cylinder clamping device that ensures a seal between the chip and its

cover as well as all of the fluid connections to the chip.

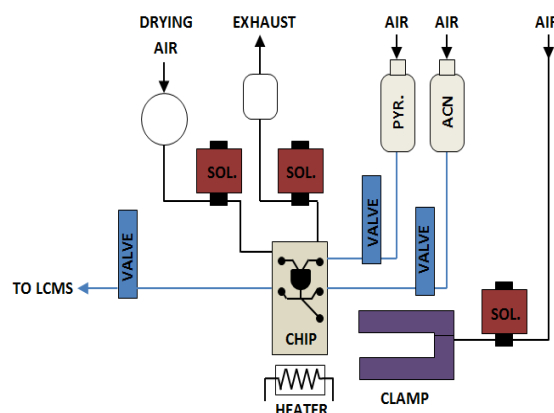


FIGURE 4. Schematic layout of extraction system

An important consideration in the design of the extraction chip is handling and loading the fiber. While the active area of the chip can be as small as 10 X 10 mm, limiting the size of the device to this would make it difficult for a forensic examiner to manipulate. Adding a handle will make the dimensions similar to a USB memory key as shown in Figure 5. This size allows a forensic examiner to easily insert the fiber into the 3x4 mm extraction chamber using a microscope and tweezers.

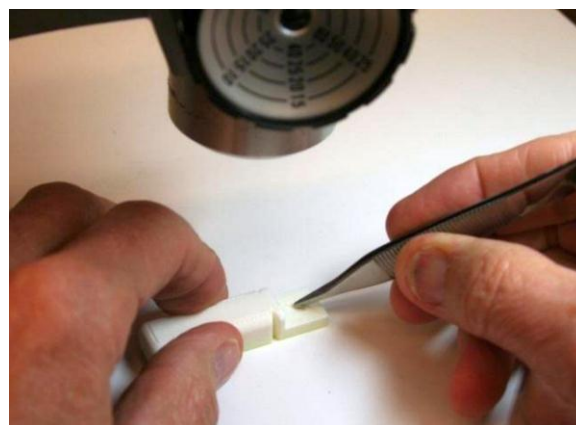


FIGURE 5. A demonstration of how a forensic examiner would place a fiber sample in a chamber on the extraction chip.

SYSTEM AUTOMATION

After placing the fiber into the chamber on the microfluidic chip, a glass cover slide is placed on it and the chip is inserted into a slot in the extraction system. The chip is then clamped into place which seals the extraction chamber and connects the chip to the fluid and air ports.

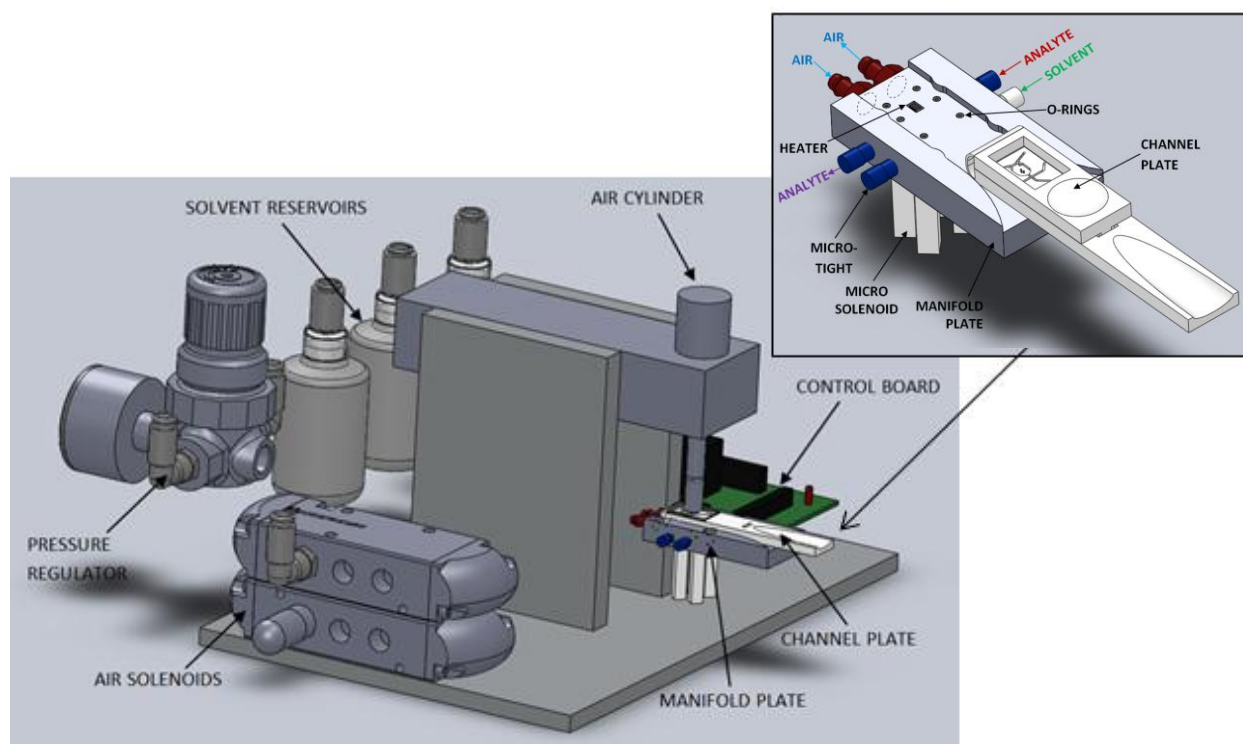


FIGURE 6. Concept for automated fiber dye extraction system. The main view on the left is the overall system including the solvent reservoirs, air supply, air cylinder for sealing the extraction chamber. The inset at the upper right is a close-up of the microfluidic chip and manifold plate.

One design that is being evaluated is shown in Figure 6. The overall view of the system is shown at the left with a close-up of the microfluidic chip being inserted into the extraction system on the right. This close-up shows the ports that connect to the chip and are sealed with o-rings. These ports include the extraction solvent, air and analyte inputs as well as the air and analyte output. Micro-solenoid valves are shown under the manifold plate and are controlled by a single board computer.

Once the microfluidic chip is inserted, it will lock into place and an air cylinder will load the glass plate against the channel plate to seal the extraction chamber. The manifold plate connects the channel plate to the fluid and air supplies. The overall system includes multiple extraction solvents, the control board, air solenoids to assist in the evaporation process and the analyte used to dissolve the dyes and provide a fluid to move them into the LC-MS for identification. The long term goal is to have the system connected to the data output of the MS so that the examiner will have quick feedback on the resulting dye identification.

CONCLUSIONS

The issues addressed during the development of this device include microfluidic flow control, chemical compatibility of chamber material, micromachining of thermoplastics, fluid interfaces, sealing the extraction chamber and realization of chemical processes such as dissolution, evaporation and mixing on the microliter scale. Calculations and tests of fluid flow control using micro valves, cryogenic machining using steel and diamond tools as well as modeling and testing of seals will be discussed along with results of LC-MS analysis.

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HIGH-PRECISION ABSOLUTE THICKNESS MEASUREMENTS OF OPTICALLY THICK COMPONENTS

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INTRODUCTION

Although traditional interferometry can be ideal for displacement metrology, it is often ill-suited for dimensional metrology such as length or thickness measurement due to fringe ambiguities [1]. A number of related technologies compromise the laser's coherence, and therefore typically the measurement precision, in exchange for the ability to measure over larger windows without ambiguity. These systems rely on large information bandwidths to synthesize full, unambiguous range profiles. For instance, white light or low-coherence interferometry offer very large bandwidths (tens or hundreds of THz) and can be used to profile surface features down to the nanometer level [2,3]. However, the short coherence length typically limits the measurement windows to tens of microns. A reference delay line in these systems can be physically scanned to enable larger effective windows, but this is typically slow, requires continual and precise calibration and monitoring of the reference delay, and is prone to failure due to the physical movement.

At Bridger Photonics, Inc. we have developed a coherent laser technology for achieving extremely high-precision, absolute length metrology over large stand-off distances, with large unambiguous range windows. To highlight these features, we measured the thickness of a 9.5 mm BK-7 window, with a precision (repeatability) of less than 1 nm, by simultaneously measuring the front and back surfaces of the window, at a stand-off distance of 12 cm.

BACKGROUND AND THEORY

Our technique, known in the laser ranging and detection (LADAR) community as frequency-modulated continuous-wave (FMCW) LADAR [4], is similar to swept frequency OCT [5] in that we chirp a narrow-linewidth laser over large bandwidths (up to 8 THz demonstrated and >15 THz possible with existing laser technology). However, our laser sources are

mode-hop-free over the entire sweep, narrower in linewidth, and we perform active laser frequency corrections in order to maintain the linearity of the chirp. These factors enable us to significantly improve the laser coherence, and therefore the measurement window, compared to OCT. A repetition rate of 5 Hz was used during the collection of the presented data. However 20 Hz has been successfully demonstrated with similar performance.

For FMCW LADAR, one copy of the chirped laser light is transmitted to the target, as shown in Figure 1. The light returning from the target (called the "return" beam) is collected and interferometrically combined with a local reference copy of the chirped light (called the "reference" beam). The combined beam is directed onto a photodetector and an RF beat note results from the time delay between the two combined copies of the chirp at frequency $f_{beat} = \kappa\tau$, where κ is the chirp rate (often in MHz/ μ s), and τ is the time delay between the two chirps. With accurate knowledge of the chirp rate and the beat note frequency, f_{beat} , the time delay can be determined. From the time delay, the range to target is given simply by $R = \tau c / 2$, where c is the speed of light.

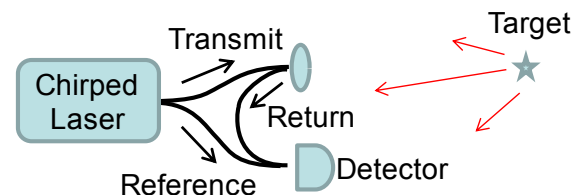


FIGURE 1. Simplified schematic of FMCW LADAR. The reference and return light are combined to generate an RF beat note measured by the detector. The frequency of the beat note indicates the target range.

The expected performance of our system can be calculated, using the following definitions

adopted by NIST. The range resolution, defined as the minimum resolvable range separation (full width at half maximum) between two targets on the same bearing, is given by $\Delta R = c/2B$, where B is the total bandwidth of the chirp. This equation makes clear the desire for larger bandwidth in order to achieve better resolution. It is only valid over measurement windows that are well within the laser coherence length. The range resolution is a very useful parameter for comparing different systems on equal footing because it does not depend on any details of the transceiver or target, such as transmitted power, target reflectivity, collection aperture, or detector noise.

Conversely, the range precision (also known as the repeatability or reproducibility) depends on the signal-to-noise (SNR) ratio and is defined as the standard deviation of a statistically significant number of range measurements of the same target. Mathematically, a lower limit for the range precision is given by the Cramér-Rao relation $\sigma \approx \Delta R / \sqrt{SNR}$. The measurement precision that a system can achieve can therefore be far better than its resolution. For the system on which we report in this abstract, we used a chirp bandwidth of 6 THz and achieved an SNR of around 80 dB. We therefore expect a resolution of 25 microns and a precision lower limit of 2.5 nm.

EXPERIMENT AND RESULTS

For this work we were interested in measuring a physically thick optical component. For the following experiment, a BK-7 wedged window with a specified physical thickness of 9.5 mm was measured. In order to improve upon previous thickness measurements, a custom optical probe was designed. The optical probe allowed for a more stable and repeatable measurement platform. The optical probe was also design to internally generate the LO copy and also includes an internal reference surface used to mitigate residual sidelobe structure on the range peaks. The window was securely mounted beneath the probe to a 2D translation stage. Figure 2 highlights these components and illustrates the final setup.

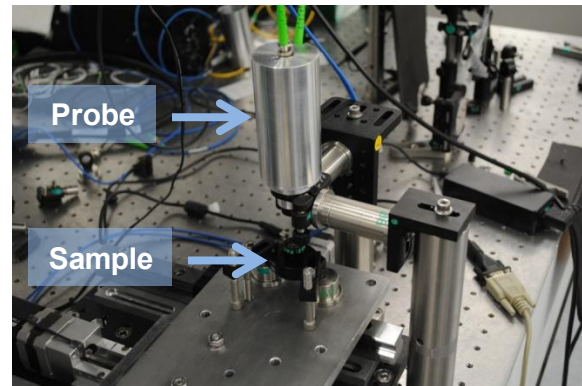


FIGURE 2. Setup used for measuring the thickness of the wedged window. The custom optical probe is vertically mounted above a 2D translation stage. The window was then securely mounted to the stage.

Figure 3 shows a zoom of the region near the front surface of the window. While the mean range to the target has been subtracted off to better illustrate the range resolution and precision, the actual stand-off distance was 12 cm. From this figure, an SNR of 80 dB is measured. The peak is also highlighted and enlarged, showing that the width of the peak is 25 microns, as expected from a bandwidth of 6 THz. By fitting a Gaussian to each of the peaks that represents each surface, we determined the center range of each surface. The 25 micron peak width coupled with an SNR of 80 dB should result in a measurement precision of approximately 2.5 nm, based on the Cramér-Rao lower bound.

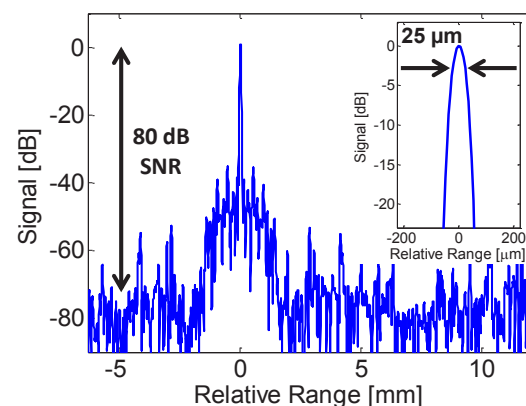


FIGURE 3. Range profile showing the front surface of a 9.5 mm BK-7 wedged window with a signal SNR of 80 dB. The transform limited range peak is highlighted and enlarged, showing a width of 25 microns.

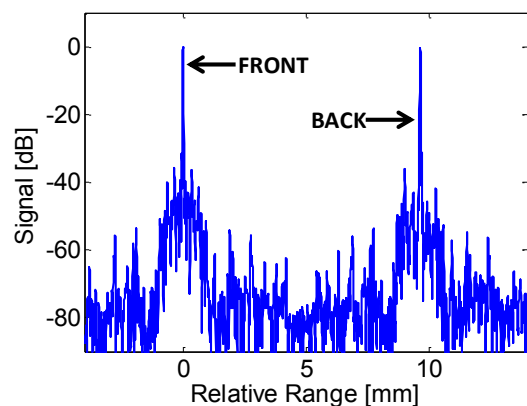


FIGURE 4. Range profile showing the front and back surfaces of the BK-7 window. The range axis has been centered on the front surface and corrected for the refractive index of BK-7 (1.5007 at a center wavelength of 1545 nm).

Figure 4 shows the range window centered on the front and back surfaces of the BK-7 window. The range axis has been referenced to the front surface and corrected for the index of refraction of BK-7. The back surface is shown to be approximately 9.65 mm from the front surface, closely matching the expected thickness of the window.

Figure 5 shows the resulting peak centers for the front (blue) and back (red) surfaces for 100 successive measurements. No averaging was performed to make these measurements. The figure also shows the thickness (black) of the target, which is obtained by taking the difference between the front and back surface ranges. The front and back surface measurements both show a precision of 4 nm and the thickness measurement shows a precision of less than 0.3 nm. As can be seen in Figure 5, the front and back surface measurements are subject to common-mode variations. By subtracting the two measurements to determine the thickness, these common-mode variations are removed and results in an improvement in measurement precision.

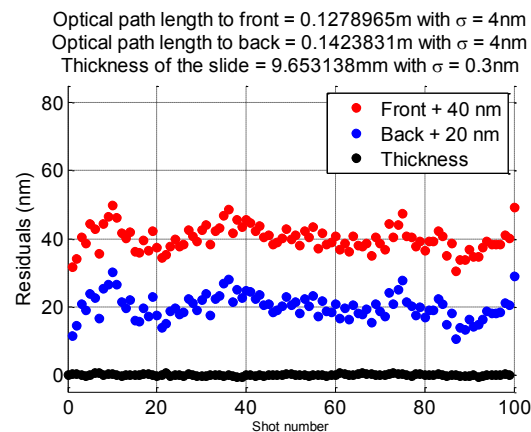


FIGURE 5. Range peak center versus measurement number for the front surface, back surface, and thickness (difference). The measurement precision for the front and back surfaces is 4 nm. The physical thickness of the window was measured to be 9.65 mm (using a refractive index of 1.5) with a measurement precision of 0.3 nm. The manufacturer thickness specification for the window is ± 0.25 mm and measurement of the window using calipers resulted in a thickness of 9.63 mm.

The measurement precisions for the front and back surfaces and the thickness of the window all show a relative precision of approximately 3 parts in 10^8 . The difference in thickness between the caliper and LADAR measurements (0.02 mm or 0.2%) can be attributed to a number of factors. The kappa value used in converting frequency to absolute length was known to 1 part in 10^4 for these measurements. The pointing accuracy directly through the window is not known. Finally, the index for BK-7 is not constant over the entire sweep bandwidth.

We are unaware of other techniques that can measure with such resolution and precision over such a large measurement window. Moreover, the measurements for both the front and back surfaces shown in Figure 5 were taken simultaneously while yielding the listed range and thickness precisions.

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INTERFEROMETRIC-TACTILE HYBRID-STYLUS FOR PRECISION FORM, ROUGHNESS AND DIAMETER MEASURING MACHINES.

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INTRODUCTION

Many precise and small automotive parts contain cylindrical or conical surfaces with form tolerances of less than 1 micrometer. Those parts are being manufactured today in high volumes. Frequently, the 3-D information on entire surfaces is needed to check the correct function of these parts. The high resolution interferometric stylus system WhitePoint allows the measurement of such high precision mechanical parts with tolerances of less than 1 μ m. It has been developed, built, tested and used under shop floor conditions already [1].

However some automotive parts contain only one or two high precision and smooth surfaces. The other surfaces have large tolerances or roughness. Often those surfaces are not cooperative with a high resolution interferometric stylus. Nevertheless it is also important to measure them or to use them for aligning of the part. For that application we further developed our interferometric measuring system and designed a new hybrid-stylus with combined interferometric and tactile probing elements using the same evaluation unit.

INTERFEROMETRICAL STYLUS SYSTEM

The optical system is based on the concept of short coherence heterodyne interferometry working with two wavelengths [2]. The prototype of the interferometer is shown in the figure 1.

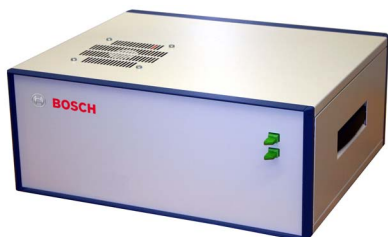


FIGURE 1. Prototype of the interferometric measuring system second generation: WPD.

The frequency shifting is realized by using one acousto-optic modulator working with a frequency of 40 MHz. The interferometer employs a broadband fiber-optic light source and works at an optical wavelength of 1,55 μ m and a synthetic wavelength of 36 μ m.

Under laboratory conditions we achieved a distance equivalent noise level of $1\sigma < 1$ nm at a sampling frequency of 10 kHz. Fig. 2 presents measurement of the distance equivalent noise of the WPD interferometrical stylus.

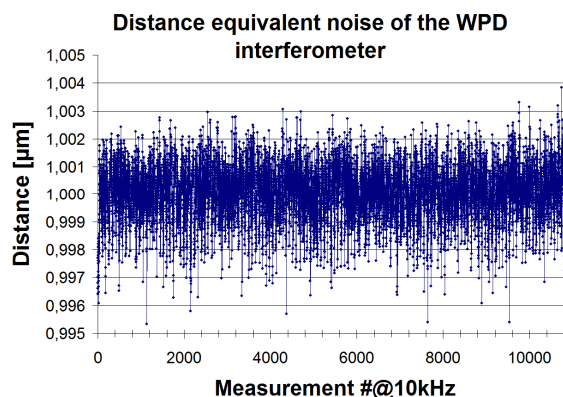


FIGURE 2. Distance equivalent noise of the interferometric measuring system: WPD.

We realized a Michelson-Type, short-coherence interferometer with two outputs. Each output could be connected through a single monomode optical fiber with a small and robust stylus.

HYBRID STYLUS

The hybrid stylus contains two interferometric probes which could be connected to the two outputs or through an optical switch to only one output of the interferometer.

The first prototype of the hybrid stylus is shown in the figure 3.

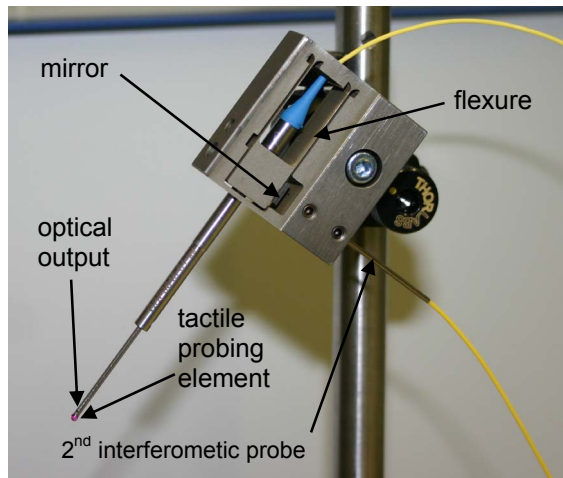


FIGURE 3. Hybrid-stylus with an interferometric and tactile probing element.

The first probe in the hybrid stylus is an interferometric one [3] containing a GRIN lens and an optical prism giving an optical output. It has a ruby ball as a tactile probing element. This stylus is placed in a special flexure which contains a mirror. The second optical probe is used to measure the displacement of the mirror and in this case to measure the displacement of the ruby ball for tactile measurements.

COMPARISON BETWEEN OPTICAL AND TACTILE OUTPUT

The comparison between optical and tactile output was performed using piezo stage with mirror (figure 4).

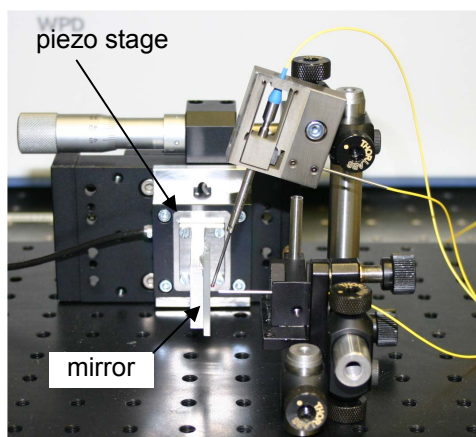


FIGURE 4. Setup with a piezo stage for tests of hybrid stylus.

The mirror was sinusoidal moved with an amplitude of $2,6\ \mu\text{m}$ and frequency of 3 Hz. The oscillation amplitude of the mirror was measured with optical output and with a tactile output how shown on the figure 4. Results of this measurement are shown in figure 5.

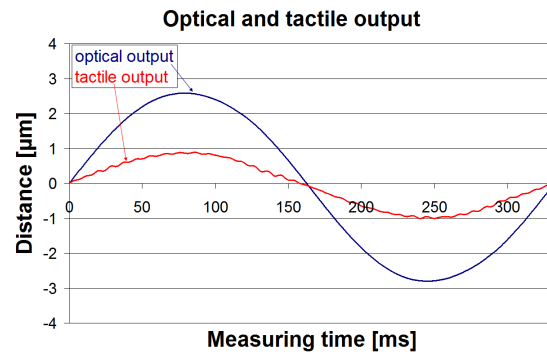


FIGURE 5. Results of comparison measurements of sinusoidal movement performed with optical and tactile output.

In this experiment we achieved transitions ratio of 2,9 between optical and tactile output.

METROLOGICAL INVESTIGATION

The metrological investigation of the hybrid stylus was performed on a ZERODUR gauge ball with a nominal roundness of 14 nm. The ball was placed on a precise air bearing rotation table with a runout error of less than 40 nm.

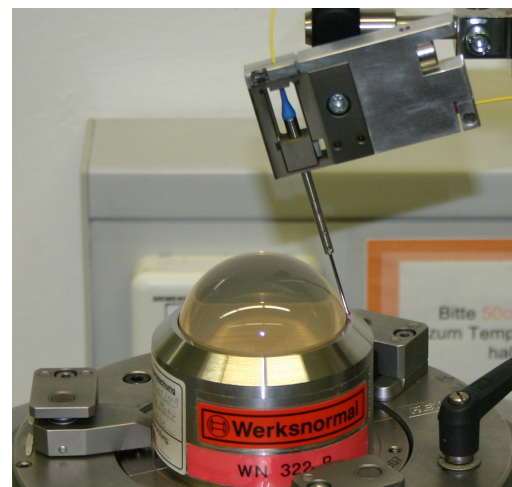


FIGURE 6. Setup with a gauge ball for testing the hybrid stylus.

In the first step we have determined the transmission ratio between displacements of the ruby ball and the mirror. This corresponds to the transmission ratio between optical and tactile output. For this experiment the ball was rotated with a small eccentricity of 2 μm . The eccentricity was measured with the optical output and the tactile probe. The transmission ratio was calculated from the results. For the given distance of 70 mm between the ruby ball and the mirror a transmission ratio of 3,7 was achieved.

In the second step we have measured the roundness on the gauge ball with the optical and tactile outputs. The measurement was performed with a rotation speed of 20 rpm and an eccentricity of 100 nm. An example of optical roundness measurement is shown in figure 7.

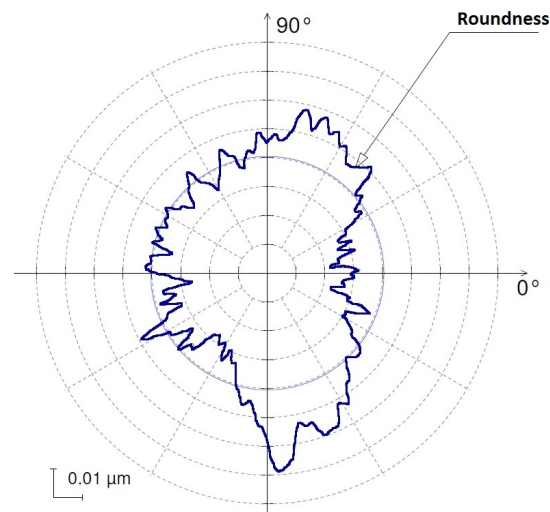


FIGURE 7. Example of optical roundness measurement on ZERODUR gauge ball.

Finally we have performed optical and tactile repeated measurement of roundness. Results of this measurement are shown in figure 8.

With 50 measurements we achieved the following results:

- optical roundness: 45 nm on average with a standard deviation of 6 nm
- tactile roundness: 53 nm on average with a standard deviation of 9 nm.

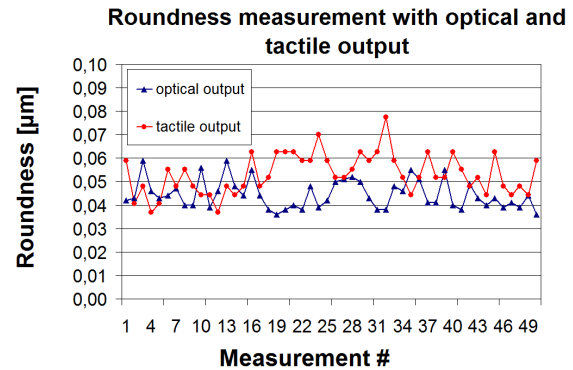


FIGURE 8. Results of the roundness measurement with optical and tactile outputs.

CONCLUSIONS

The difference of the achieved transmission ratio between optical and tactile output in our two experiments indicates non linear characteristic of the tactile stylus. The new flexure will be designed and constructed. The goal is to get the better linearity and to reduce the probing force.

The constructed flexture allows bidirectional backlash-free tactile measurements. That feature is important for precise diameter measurement. We already developed a bidirectional, non contact interferometrical stylus and are now working on a hybrid stylus for diameter measurements.

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SEISMIC PLATFORM INTERFEROMETER – AN AUXILIARY SENSOR FOR USE IN ADVANCED LIGO

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INTRODUCTION

We are trying to measure gravitational waves from astrophysical sources. The Laser Interferometer Gravitational-wave Observatory (LIGO) is a large project designed to detect directly gravitational waves through the use of three, long-baseline interferometers and is funded by the National Science Foundation (NSF).

To improve the estimated event rate by a factor of about 1000, the LIGO project is currently installing the Advanced LIGO (aLIGO) upgrade increasing the sensitivity of the detectors [1], [2], and [3]. Commissioning work for aLIGO has already started with the upgrade scheduled to come on-line in 2015.

Several significant changes are being made to improve the detectors' performance. One of these changes is the installation of an upgraded seismic isolation system. This is necessary to attenuate the transmission of ground motion to the suspended optics by as much as 10 orders of magnitude at 10 Hz. Seismic isolation starts with quiet Hydraulic External Pre-Isolators (HEPI) outside the vacuum system, Internal Seismic Isolation (ISI) platforms inside the vacuum envelope, and then up to four stages of pendulums down to the final proof mass optic.

As part of aLIGO, 30 six degrees-of-freedom actively controlled, Internal Seismic Isolation (ISI) platforms are being installed. The platforms are actively controlled relative to inertial space utilizing seismometers. At low frequencies, however, the horizontal feedback seismometers have difficulty distinguishing horizontal accelerations from

the component of gravity (g) due to tilt. The ratio of a horizontal seismometer's sensitivity to rotation (signal per radian of angle) to the sensitivity to translation (signal per m) at a particular frequency (denoted ω in rad/s) is represented by:

$$\frac{\text{rotation response}}{\text{translation response}} = \frac{-g}{\omega^2} \quad (1)$$

This tilt-horizontal coupling is one of the factors limiting the low frequency performance of the ISI system [4].

Several options could address the problems caused by the tilt-horizontal coupling in the feedback inertial sensors. We decided to use an auxiliary Seismic Platform Interferometer (SPI) sensor, a concept originally suggested by Robertson [5] to measure the differential displacement between adjacent platforms for low frequency control instead of the tilt-dominated local inertial sensors. Below the first natural frequency of the pendulums (about 0.4 Hz), absolute platform displacement is permissible if the differential displacement of the platforms is minimized. Therefore, if the differential motion of the pendulums' supports is controlled, then the differential motion of the mirrors will also be reduced to an acceptable level, even if a group of platforms are moving in the absolute sense at very low frequencies.

The SPI is designed to measure and generate control signals for the differential length, pitch, and yaw between platforms. Yaw (rotation about the vertical z axis) is critical to the pointing and alignment of the pendulums, optics, and subse-

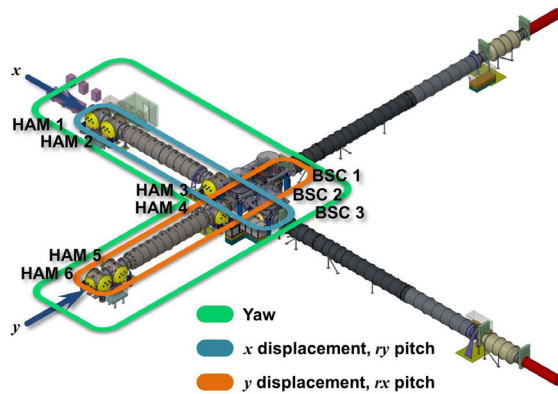


FIGURE 1. aLIGO corner station Internal Seismic Isolation (ISI) platforms and the required control dimensions for the SPI in order to counteract the tilt-horizontal coupling in the inertial sensors.

quent laser beams. Pitch (about the x axis) is necessary because the attachment points for the pendulums are not co-located with the SPI sensor. Figure 1 shows the active ISI platforms (denoted as BSC or HAM) in the aLIGO corner station with the corresponding control dimensions.

The SPI control is envisioned to take place within a target frequency range from 5×10^{-3} to 5 Hz. This is below the science band of the detector (from about 10 Hz to several kHz) in order to allow the aLIGO optical proof masses freedom to respond to a passing gravitational wave. The target resolution for the SPI is documented in Table 1.

TABLE 1. SPI target resolution for length sensing $\tilde{x}$ in $m/\sqrt{Hz}$ and angular pitch and yaw sensing $\tilde{\theta}$ in $rad/\sqrt{Hz}$ as a function of frequency.

Freq. Hz	0.01	0.1	1
$\tilde{x}$ Target $\frac{m}{\sqrt{Hz}}$	$4 \cdot 10^{-5}$	$4 \cdot 10^{-7}$	$4 \cdot 10^{-10}$
$\tilde{\theta}$ Target $\frac{rad}{\sqrt{Hz}}$	$4 \cdot 10^{-5}$	$4 \cdot 10^{-7}$	$4 \cdot 10^{-10}$

EXPERIMENTAL SETUP

The Stanford Engineering Test Facility (ETF) currently operates an SPI prototype. Length sensing is accomplished by a Mach-Zehnder interferometer while a paired optical lever measures pitch and yaw.

The SPI prototype consists of a 1533 nm fiber coupled laser and isolator (designated ISO) outside of the vacuum system (Figure 2). The light is introduced into the vacuum chamber through polarization-maintaining fiber and an

ultra-high vacuum compatible feedthrough where it is launched from the fiber through a free-space coupler (FSC) on the isolation platform. By mounting the FSC to the isolation platform, the emitted light beam is optically anchored to the host platform. The light then travels through a half-wave plate (HW) and a polarizing beam splitter (PBS) to strip stray polarization from the launched light. The beam is then divided into the measurement and reference arms by the first beamsplitter (NPBS). At the remote platform (located 8.9 m away in our experiments), the light reflects back to the host platform from a single mirror. Before most of the return beam is steered (SM) onto the second beamsplitter, part of it is monitored on a quad photo-detector (QPD) which measures the differential pitch and yaw of the two platforms. On the second beamsplitter (NPBS), light from the short reference arm and the return light from the remote platform are interfered and recorded by a balanced photo-detector pair.

The SPI was designed with two operational modes. One mode, relying on mid-fringe locking, locks the reference arm through the control of a piezoelectric transducer (PZT) to the sensing arm. The control signal necessary to keep the interferometer locked on the mid-fringe is then proportional to the differential displacement between the two arms. Locking on the mid-fringe reduces the coupling of the laser source intensity noise into the measurement by maintaining a constant nulled value on the balanced photo-detector. Limitations in this mode include a range limited by the PZT range and hysteresis of the PZT. The second mode, relying on fringe counting, modulates the light in the short, reference arm. The interference signal from the photo-detectors is then demodulated through the use of a lock-in amplifier. While fringe counting is not limited in range, the measurement velocity is limited by the modulation rate. Amplitude noise of the laser could also affect the signal in the fringe counting mode because the signals on the photo-detectors are not necessarily balanced.

Because the inertial sensors already on the platforms are used for local control, two Geotech GS-13 witness seismometers were installed to validate the SPI prototype. While these witness seismometers are susceptible to the component of gravity coupling due to platform tilt, they are the most sensitive instruments available to us, especially in the 1 to 10 Hz frequency band.

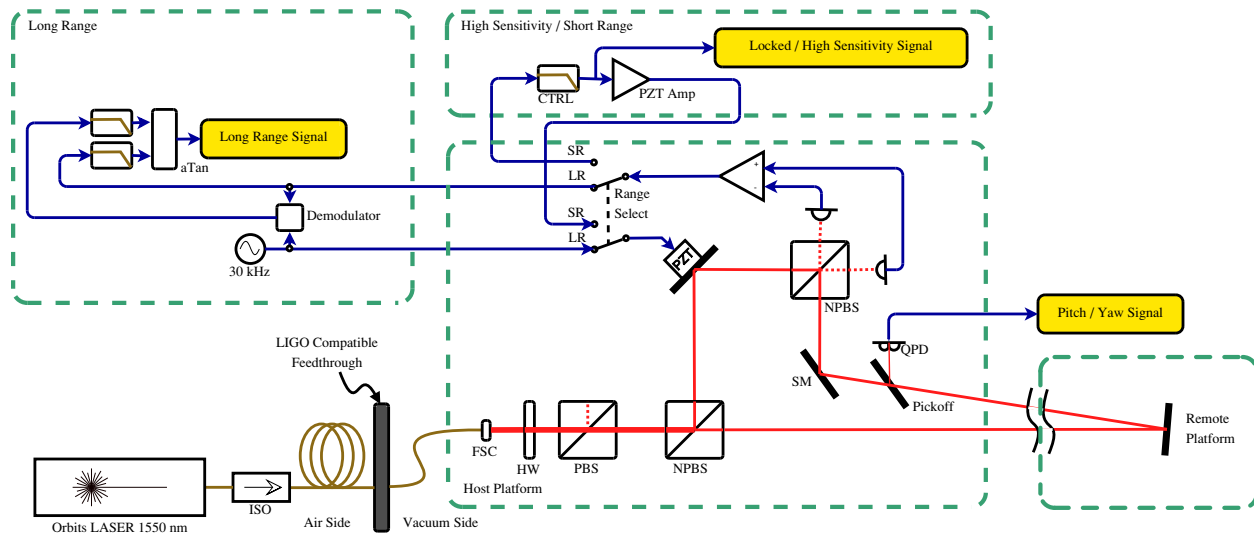


FIGURE 2. Schematic diagram of the SPI prototype currently operating between platforms in the Stanford Engineering Test Facility.

NOISE CHARACTERIZATION

Since SPI incorporates unbalanced arms, frequency noise of the laser source becomes indistinguishable from differential length between platforms. To quantify this, tests were conducted to characterize the laser used and are shown in Figure 3. The frequency noise is already below the requirement but it is thought that some of this noise is caused by thermal coupling.

The laser frequency noise is the limiting source in the target frequency range of the SPI. Other noise sources investigated and quantified include the computer ADC and DAC systems, the photodetector electronics noise, shot noise, and the PZT amplifier noise.

CONTROL OF PLATFORMS

The SPI prototype signal was then used to control the differential motion of two actively controlled isolation platforms in the ETF. These platforms were separated by a distance of 8.9 m with one two stage system and the other only a single active stage. The two stage platform served as the host platform.

The platforms were then locked together by the SPI signal in differential displacement, pitch, and yaw through control of the remote platform. This configuration allowed the remote platform to be actively driven closer to the higher performance levels of the host platform. The differential displacement motion recorded on the witness GS-13

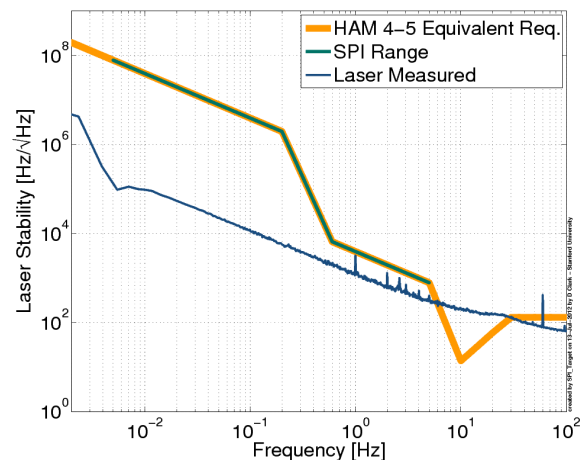


FIGURE 3. Frequency noise of the laser source as measured by displacement of a fixed 3.01 m mismatched arm on top of the isolation platform with the best linear fit removed. The noise is compared to the target motion projected from a system with a 27 m offset between platforms. This is the longest distance between platforms in the LIGO corner station where the SPI will operate and represents the most stringent operating location.

seismometers showed the SPI achieved an order of magnitude reduction in motion between the frequencies of 50 mHz to 5 Hz (Figure 4). While the motion reduction was significant, it did not meet the reduced motion predicted by the control loop

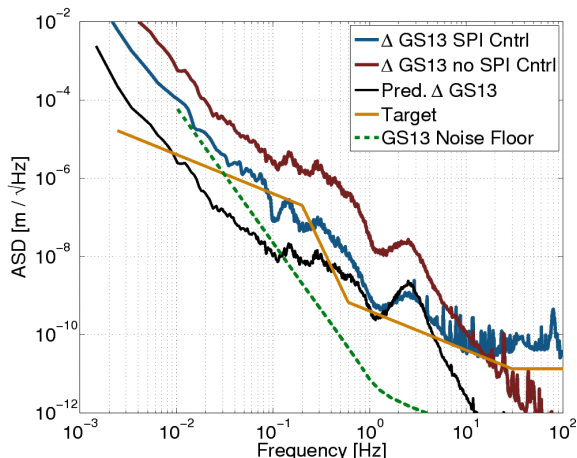


FIGURE 4. SPI controlled reduction in the differential motion between two active platforms as measured by witness seismometers at frequencies below 15 Hz. The measured motion (blue curve) is not as low as expected (black curve) and is under investigation with one possibility being tilt-coupling in the sensors.

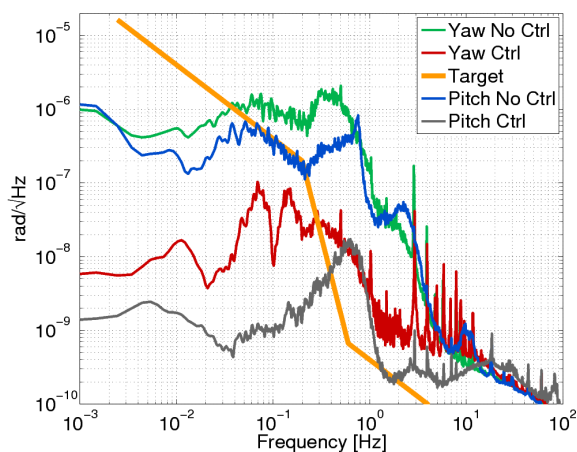


FIGURE 5. SPI controlled reduction in the differential pitch and yaw motion between two active platforms as measured by the in-loop SPI sensor.

gain. The cause is still under investigation but one possibility is witness sensor errors due to tilt-horizontal coupling. Also observed is excess motion induced at frequencies above 15 Hz caused by high loop gain close to the SPI noise floor at these frequencies. In practice, the SPI controller should be designed to relinquish control authority above 5 Hz preventing SPI sensor noise from being introduced and to allow the ISI platforms to move independently at high frequency.

While no witness instruments were installed to record differential pitch and yaw, the in-loop SPI signal also displayed high reductions in motion while under control (Figure 5).

CONCLUDING REMARKS

The SPI prototype demonstration successfully measured and controlled the differential displacement, pitch, and yaw between two active seismic isolation platforms. The SPI therefore provides a possible solution to induced, unwanted motion at low frequency caused by the tilt-horizontal coupling in the feedback seismometers of the active ISI platforms in Advanced LIGO.

ACKNOWLEDGMENTS

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MINIATURE, FIBER-COUPLED 3-DOF INTERFEROMETER FOR PRECISION MICRO-MOTION STAGE METROLOGY

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INTRODUCTION

The objective of this research is to demonstrate a compact, fiber-delivered displacement measuring interferometer which can be used to simultaneously calibrate the linear motion and rotational errors of a precision nanopositioning stage. Heterodyne interferometry is ideal for this type of high resolution stage feedback sensing because of its high dynamic range, high signal-to-noise ratio, and direct traceability to the meter.

There are a variety of benefits to fiber-delivered interferometers for stage metrology. First, the interferometer alignment is decoupled from the laser alignment. The overall setup can be divided into subsystems to more easily track down and rectify misalignments. Simplified alignment and division into subsystems increases the stability of the structure because the metrology footprint is significantly reduced. Many beam routing optics can be eliminated and the heterodyne laser (a heat source) can also be isolated from the stage measurement site.

In this proceedings, we present ongoing work towards a compact, three degree-of-freedom fiber-delivered heterodyne interferometer. Our novel interferometer will utilize differential wavefront sensing to measure target mirror displacement, pitch, and yaw [1,2]. Differential wavefront sensing uses a quadrant detector to measure four spatially separated interference signals within a single optical interference beam. Based on the geometry of the detector and the interference phase in each quadrant, the displacement and changes in target pitch and yaw can be measured.

ERROR CONSIDERATIONS

The three main error sources that limit the performance of a displacement interferometry system are the laser frequency stability [3], refractive index fluctuations in non-common optical paths [4], and periodic nonlinearity in the measured phase due to source mixing, manufacturing tolerances, and imperfect alignment [5-7].

For micro- and nano-positioning stages, the measurement uncertainty scales with the optical path difference between the measurement and reference arms in the main interference signal. Provided the nominal difference between the measurement and reference arms is sufficiently small (<1 mm), the uncertainty contributions from laser frequency instability and refractive index instability can be nanometers or less. Thus, the limiting uncertainty contributor for the interference measurement is from periodic nonlinearity due to source mixing, imperfect coatings, and alignment.

Previous research has shown that it is possible to build heterodyne interferometers with spatially separated beams to limit or eliminate periodic nonlinearity (e.g. [8-11]), including some configurations that are suitable for employing fiber-delivery to the interferometer [12]. This limits the coupling between free-space beam delivery components for multi-axis systems.

Fiber delivered interferometers have an additional drawback which is the input fiber induces artificial Doppler shifts to the optical reference and measurement signals. Traditional heterodyne interferometry has a nominally fixed optical reference signal and a measurement signal that fluctuates based on the optical

Doppler shift induced by the moving measurement target. If the measurement target is not moving, then both the reference and measurement signals are nominally equal and constant. Fiber delivered interferometers have a continuously changing heterodyne frequency due to continuous fluctuations in temperature, stress, and vibrations on the input fibers. These perturbations can be equal for both the reference and measurement interference signals in certain interferometer configurations but it places additional design considerations on the phase detection algorithm.

SYSTEM SETUP

A schematic of our compact, fiber-delivered interferometer is shown in Figure 1. The laser output beam is split and directed towards two acousto-optic modulators (AOMs) which create the split frequency needed to obtain frequency modulated measurements. The AOMs eventually produce a nominally 70 kHz heterodyne frequency. This split frequency was chosen because quadrant detectors are commercially available at this bandwidth and we did not anticipate any issues with speed limitations for our calibration parameters. However, the phasemeter system developed for this work uses 50 MSPS ADCs, so in principle, we could switch to a higher heterodyne frequency provided the AOMs were chosen appropriately.

After passing through the AOMs, the first order upshifted beams are then both fiber-coupled into polarization maintaining (PM) fibers. As previously stated, both fibers are prone to stress induced phase shifts as a result of mechanical and thermal stresses. The phase shifts, denoted by Θ_1 and Θ_2 , can be kept nominally equal if the

fibers are tied together, however, the fiber-induced phase shifts will never be the same. In terms of alignment, the first nonpolarizing beamsplitter, both AOMs, and both input fiber couplers are completely isolated from the rest of the interferometer. Therefore, a misalignment with an AOM, for example, will not necessarily propagate through the rest of the interferometer and can be corrected independently from interferometer alignment.

Both beams are launched out of their respective PM fibers into the interferometer through two fiber collimators which are spatially separated to minimize the effects of frequency mixing [12] and the resulting periodic nonlinearity [5]. Our novel interferometer consists of two non-polarizing beamsplitters, which avoids additional frequency mixing from polarization-dependent components. The quadrant detector outputs irradiance values from four photodetectors which are then post-processed. Pitch and yaw measurements are obtained using weighted phase averages over symmetrically adjacent pairings of detector elements while overall displacement is a result of the total average phase over all four detectors.

One of the additional benefits of this interferometer is pinpoint calibration on a small target within the work volume of the micropositioner. A schematic of this is shown in Figure 2. This enables the calibration of displacement, pitch, and yaw at a point off-axis from the actuator and sensor, reducing the effects of Abbe error. This can enable an accurate calibration of the displacement at a specific point-of-interest, even though the sensor is not located on the same Abbe axis.

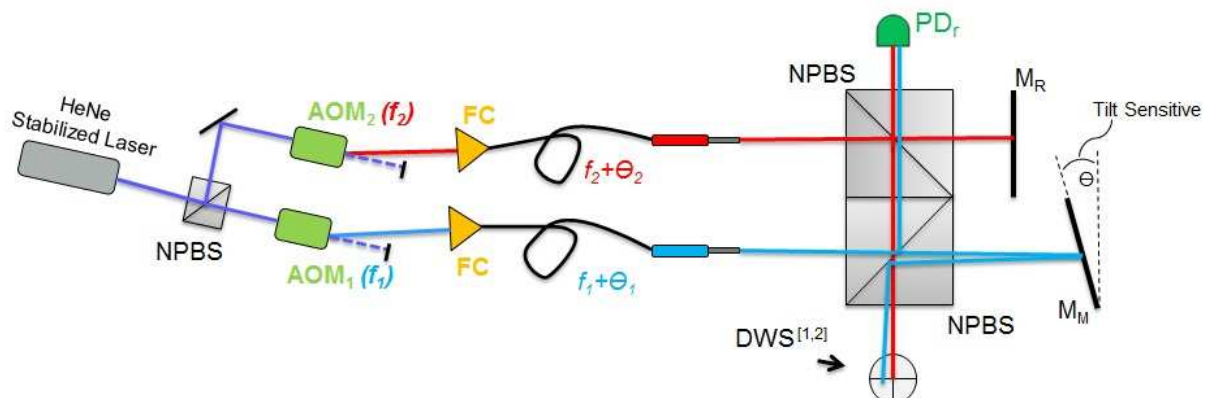


FIGURE 1. Full tip/tilt sensitive displacement interferometer set-up. (NPBS: non-polarizing beam splitter, AOM: acousto-optic modulator, FC: fiber coupler, PD: photodetector, M: mirror, DWS: differential wavefront sensing)

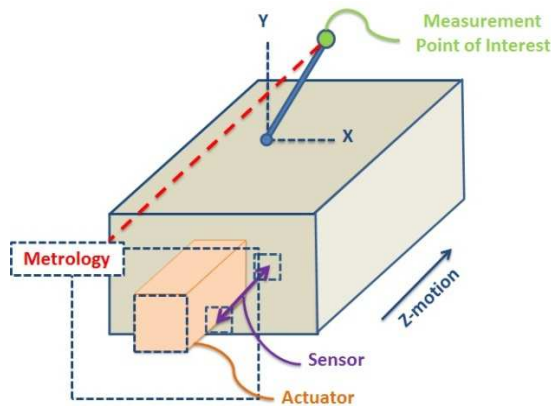


Figure 2. Schematic of measuring a particular point of interest off-axis from the Abbe position.

The stage that was being calibrated is an InSituTec IT.PS.X20.TI.CS.RZ compact piezoelectric stage with 19 μm of motion and sub-50 μm positioning resolution. Figure 3 shows a picture of the stage and Figure 4 shows an initial setup of our interferometer calibrating a different piezo stage in preliminary testing.

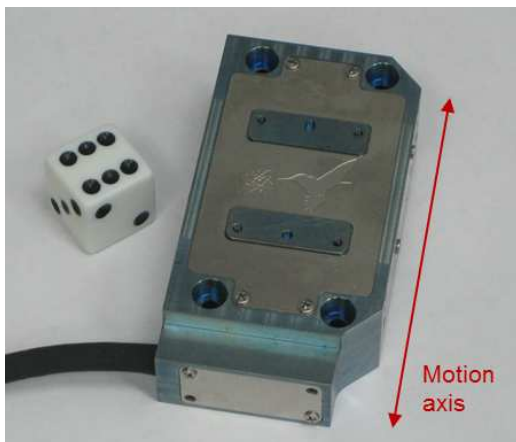


Figure 3. InSituTec X20 piezoelectric stage.

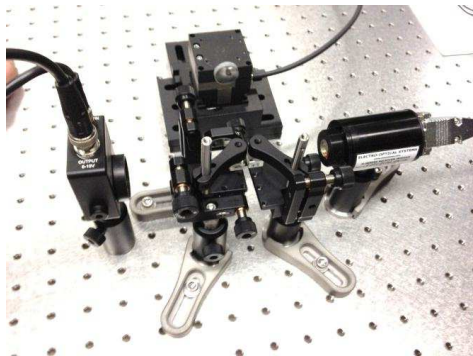


FIGURE 4. Initial setup of the interferometer calibrating a different piezo stage in preliminary testing.

RESULTS

Initial measurements were made by slowly scanning the piezo stage from Figure 4 in closed loop control after applying a custom linearization algorithm to demonstrate displacement, pitch, and yaw sensitivity and the results are shown in Figure 5. The measured rotational errors for a scan in the Z-direction are within the specifications of the piezo stage.

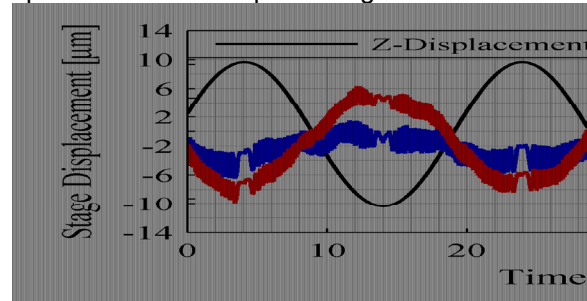


FIGURE 5. Initial measurement of the piezo stage in open loop scanning; $\sim 20 \mu\text{m}$, pk-pk of Z-displacement with $\sim 0.9 \mu\text{rad}$ and $\sim 0.4 \mu\text{rad}$ of pitch and yaw motion, respectively.

Figure 6 shows the results of a stationary test in closed-loop control to examine the noise level of the system. The linear displacement noise was approximately 0.6 nm, pk-pk with a standard deviation of 0.107 nm while the rotational noise was approximately 0.25 μrad , pk-pk in yaw and 0.2 μrad , pk-pk in pitch. These results were obtained using a sampling frequency of 1 kHz and a final low pass filter at 300 Hz.

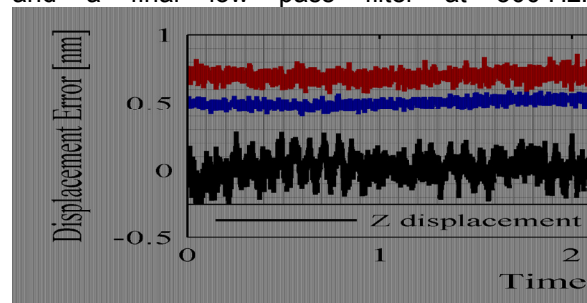


FIGURE 6. Preliminary noise measurements with 300 Hz low-pass filter implemented. Stage was held at a constant position. Data axes have been offset for clarity.

The results from Figure 6 were analyzed in the Fourier domain to assess the noise spectrum, which is shown in Figure 7. Large spikes below 100 Hz are attributed to known ground vibrations. The vertical dotted line at 300 Hz signifies the final low pass filter on the data.

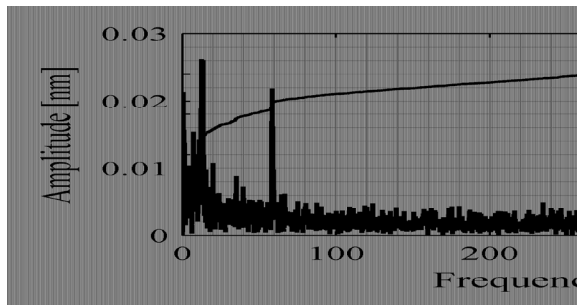


FIGURE 7. Fourier noise spectrum of Z-axis displacement. Spikes below 100 Hz are attributed to known ground vibrations. The 300 Hz low-pass filter which was implemented to obtain results of Figure 6 is drawn in red for reference.

After noise measurements were recorded with the stage held in closed-loop control, a series of random steps were measured to assess the performance of the linearization algorithm. Fifty random steps at random locations within the 19 μm work volume were performed and the difference between the internal sensor and the interferometer was calculated after settling. The error is calculated from a single measurement point (no averaging). Figure 8 shows the result of closed-loop control random steps. Clearly, there are no correlations relating to the stage position nor the step size, signifying the linearization algorithm is not showing bias.

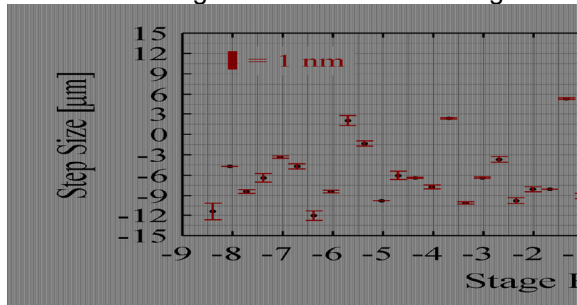


FIGURE 8. Error after settling for 50 random steps of varying sizes and varying locations throughout the work volume. The x-axis designates the initial starting position of the stage while the y-axis designates the controlled step size from that initial location. Red error bars represent the interferometer's measurement error compared to the stage's built-in capacitance sensor. The standard deviation of the error is 0.385 nm over all 50 random steps.

It is evident that, under closed loop control, the interferometer is capable of sub-nanometer accuracy. Furthermore, step response repeatability was independent of initial stage

position and step size, signifying a robust controller and an interferometer capable of high-resolution closed-loop control.

CONCLUSIONS AND FUTURE WORK

We have presented ongoing work towards the development of a fiber-coupled multi-degree of freedom displacement interferometer for high precision stage metrology. The interferometer has demonstrated a Z-displacement noise level of 0.1 nm and rotational noise levels of $\sim 0.1 \mu\text{rad}$. In a random step test, the X20 piezo stage showed repeatability below 0.4 nm over the full working volume.

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A 6-Axis Position Sensor

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INTRODUCTION

Most positioning applications require more than one axis of motion. For example, in machining applications, what is most interesting is the relative position (both translations X/Y/Z and rotations A/B/C) between the part to be machined and the tool. Existing solutions rely mainly on stacking multiple single-axis sensors to synthesize the relative position between two rigid bodies, over a very long structural loop. The problem is that bearing motion errors, structural deformation and/or expansion due to mechanical and thermal load can significantly degrade the fidelity of the synthesized position. A separate load-free, thermally-stable metrology frame is implemented in extremely high-precision applications like large optics diamond turn machining (LODTM) and lithography, but remains impractical in most applications. Industrial robots face the same problem but even more severely. Positioning an end effector of a robot arm with only joint sensors typically limits accuracy to the millimeter level, despite achievable repeatability of tens of microns. Ideally, the relative position between two rigid bodies should be measured directly in all 6 degrees of freedom (DOF). In reality, most position sensors are 1 or 2 DOF with very strict motion requirement on all other DOFs. 6-axis inertial sensors are readily available, but don't improve position accuracy due to integration drift.

One class of technologies that can potentially measure all 6-axis motion is a vision-based metrology system. By processing the images acquired on multiple cameras, translation and rotation information can be directly computed. Wii™ remote control and Xbox Kinect™ have had great impact. Recent developments in optical sensor technology make vision-based metrology an attractive solution for industry motion measurement. Compared with other solutions, vision-based metrology has the advantages of non-contact sensing, large measurement volume, and simultaneous measurement of multi-DOFs. In order to achieve

precise and reliable spatial measurements, camera calibration has always been a necessary prerequisite. Various calibration methods have been reported based on different camera models, calibration algorithms and experimental setups. Eleven intermediate parameters in a direct linear transformation method (DLT) were utilized to represent a linear transformation between the reference 3D coordinates and the 2D image coordinates [1]. However, lens distortions were not considered in the DLT method. A camera model that incorporates lens distortions was reported in [2]. Nonlinear optimization is required to estimate camera model parameters when lens distortions are considered. Several estimation algorithms were reported in the literature to solve the model parameters [3] [4] [5]. These methods generally involved multiple steps: a linear method was utilized to estimate most of the model parameters or their initial values, and nonlinear optimizations were then applied to obtain the final model parameters.

The accuracy of a camera calibration method does not only rely on the camera model and the estimation algorithm of model parameters, but also requires reliable calibration test points that provide accurate calibration information. In prior art, calibration test points were generated by printed patterns, patterns made by photolithography process (e.g., [4]) or other engineered artifacts (e.g., [3] [5]). Multiple test points can be obtained in a single camera frame by using these calibration patterns. However, the accuracy of calibration patterns generated by these techniques is limited by the manufacturing process, and at the same time it is difficult to measure these patterns with a high-accuracy measurement machine such as CMM. In addition, non-coplanar calibration test points were usually generated by moving a coplanar pattern with a motion stage [3] [4], therefore the accuracy of the motion stage limited the accuracy of test points.

Coordinate measurement machines (CMM) can be employed to generate accurate camera calibration test points. Meanwhile, flexible patterns of calibration test points can be generated through programming on CMM. Shen and Menq utilized the CMM to perform an automatic camera calibration [6]. The tip of the CMM touch probe was employed as the moving target to generate test points. To locate the center of the tip in the image plane, additional image processing techniques, such as feature extraction and fitting, are required. Wiles et al presented an experimental setup for camera calibration using an LED marker and a CMM [7]. The active LED marker is moved by a CMM to generate calibration test points. Accurate 2D locations of test points in image coordinates can be obtained through sub-pixel target location methods, where the LED marker is approximated as a point light source. However, a low 3D position accuracy of 0.35mm RMS was achieved by Wiles' experimental setup. However, existing vision-based position sensors are either low speed or low precision. For industrial applications, both sampling rate and precision of position feedback sensor are important. This paper presents a high accuracy camera calibration method using a CMM and active LED marker. The achievable accuracy of this 6-axis sensor is demonstrated by experiment.

CAMERA CALIBRATION THEORY

Generally, the camera model can be described as a linear transformation combined with nonlinear distortions, presenting a transformation from a 3D world coordinates to a 2D pixel coordinates. As shown in Figure 1, the relationship between the world coordinates $P_W(X_W, Y_W, Z_W)$ and non-distorted image coordinates $P_I(X_I, Y_I)$ is given by [Abdel-Aziz et al., 1971]:

$$\begin{bmatrix} X_C \\ Y_C \\ Z_C \end{bmatrix} = \begin{bmatrix} r_{11} & r_{12} & r_{13} \\ r_{21} & r_{22} & r_{23} \\ r_{31} & r_{32} & r_{33} \end{bmatrix} \begin{bmatrix} X_W - T_x \\ Y_W - T_y \\ Z_W - T_z \end{bmatrix} \quad (1)$$

$$\begin{bmatrix} X_I \\ Y_I \\ 1 \end{bmatrix} = \frac{1}{Z_C} \begin{bmatrix} -d & 0 & 0 \\ 0 & -d & 0 \\ 0 & 0 & 1 \end{bmatrix} \begin{bmatrix} X_C \\ Y_C \\ Z_C \end{bmatrix} \quad (2)$$

where $[r_{ij}]$ is the rotation matrix (defined by three Euler angles) that describes the camera orientation relative to the world coordinates, $[T_x, T_y, T_z]$ is a translation vector that describes the relative camera position in world

coordinates, and d is the principal distance between the optical center $[X_C, Y_C, Z_C]$ and the

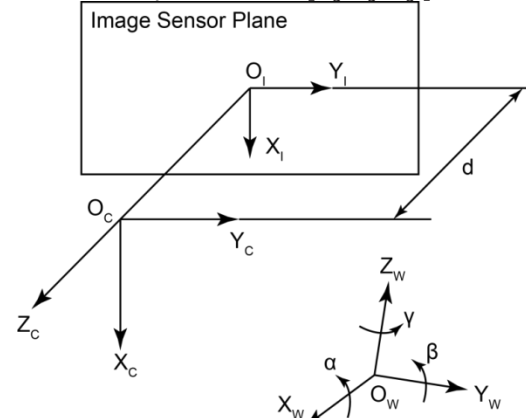


Figure 1: Transformation from world coordinates to non-distorted image coordinates

image sensor plane. Combining Eqn. (1) and Eqn. (2) gives:

$$\begin{aligned} X_I &= -d \frac{r_{11}(X_W - T_x) + r_{12}(Y_W - T_y) + r_{13}(Z_W - T_z)}{r_{31}(X_W - T_x) + r_{32}(Y_W - T_y) + r_{33}(Z_W - T_z)} \\ Y_I &= -d \frac{r_{21}(X_W - T_x) + r_{22}(Y_W - T_y) + r_{23}(Z_W - T_z)}{r_{31}(X_W - T_x) + r_{32}(Y_W - T_y) + r_{33}(Z_W - T_z)} \end{aligned} \quad (3)$$

Digital image sensors present the location information of an image array in units of pixels. The relationship between the non-distorted image coordinates and the pixel coordinates is governed by

$$\begin{aligned} u_I &= u_0 + S_x X_I \\ v_I &= v_0 + S_y Y_I \end{aligned} \quad (4)$$

where $[u_0, v_0]$ is the principal point of the image sensor plane, and S_x and S_y are the scaling factors. Denote that $f_x = dS_x$ and $f_y = dS_y$, and it is derived from Eqn. (3) and Eqn. (4) that

$$\begin{aligned} u_I &= u_0 - f_x \frac{r_{11}(X_W - T_x) + r_{12}(Y_W - T_y) + r_{13}(Z_W - T_z)}{r_{31}(X_W - T_x) + r_{32}(Y_W - T_y) + r_{33}(Z_W - T_z)} \\ v_I &= v_0 - f_y \frac{r_{21}(X_W - T_x) + r_{22}(Y_W - T_y) + r_{23}(Z_W - T_z)}{r_{31}(X_W - T_x) + r_{32}(Y_W - T_y) + r_{33}(Z_W - T_z)} \end{aligned} \quad (5)$$

In real applications, the perspective projection is a basis that is extended with some corrections for systematically distorted image coordinates [4] [5]. The distorted image coordinates $[u_D, v_D]$ is expressed as

$$\begin{aligned} u_D &= u_I + \delta'_u(u_D, v_D) \\ v_D &= v_I + \delta'_v(u_D, v_D) \end{aligned} \quad (6)$$

The lens distortion is expressed as:

$$\begin{aligned}\delta'_u(u_D, v_D) &= K'_1 u_D (u_D^2 + v_D^2) + K'_2 u_D (u_D^2 + v_D^2)^2 \\ &\quad + P'_1 (3u_D^2 + v_D^2) + 2P'_2 u_D v_D + S'_1 (u_D^2 + v_D^2) \quad (7) \\ \delta'_v(u_D, v_D) &= K'_1 v_D (u_D^2 + v_D^2) + K'_2 v_D (u_D^2 + v_D^2)^2 \\ &\quad + P'_2 (u_D^2 + 3v_D^2) + 2P'_1 u_D v_D + S'_2 (u_D^2 + v_D^2)\end{aligned}$$

where K'_1 and K'_2 represent curvature distortion, P'_1 and P'_2 are related to decentering distortion, and S'_1 and S'_2 are thin prism distortion coefficients [4].

Based on the camera model described above, the camera calibration problem can be stated as: given known test points (X_W, Y_W, Z_W) in world coordinates and their corresponding image locations (u_D, v_D) in image sensor coordinates, calculate the optimal model parameters. In order to achieve accurate model parameters, full-scale nonlinear optimization is employed. Model parameters are divided into two groups: non-distortion parameters, including camera orientation and location, image sensor scaling factors, sensor to lens distance, image sensor offsets relative to the optical axis, and diverse lens distortion parameters. The initial values of non-distortion parameters are obtained through the DLT method, and the initial values for distortion parameters are all set to zero [4] [5]. The Levenberg-Marquardt optimization algorithm is employed to obtain the optimal estimation of all unknown parameters.

EXPERIMENTS

The camera system to be calibrated is a high-speed stereo-vision system for position sensing developed in the Precision Mechatronics Laboratory of UBC. This vision system is designed based on two image sensors each with a resolution of 2048×2048 pixels. To achieve large measurement volume, wide-angle lenses with 25.7 mm focal lengths are utilized.

Figure 2 illustrates the experimental setup based on the CMM. Eight hundred test points are evenly distributed in a volume of 400mm by 400mm by 15mm in four planes with normal direction in Z_W . An active LED marker is moved by the CMM probe holder in the grid of reference positions. At each reference position, the 3D position of the marker measured by the CMM and its 2D position in each of the two image sensors are recorded. For the purpose of minimizing the side-effect of ambient light, the LED marker works in the infrared region. The active LED marker is driven by a current source

circuit. The CMM used in the experiments is features a working range of 705mm by 1005 mm by 605mm, a spatial resolution of 0.1 μm and maximum permissible error of 3.0 μm in the calibration volume.

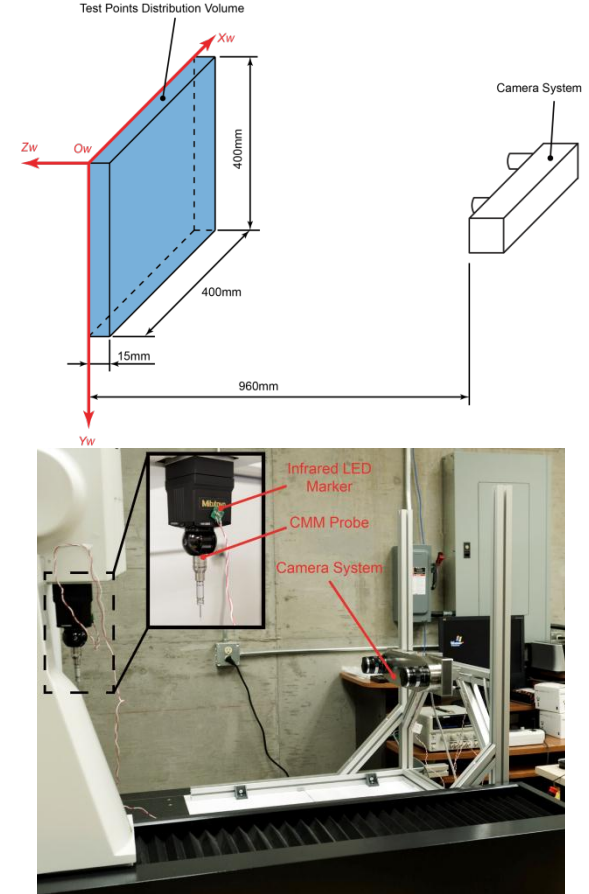


Figure 2: Illustration of camera calibration setup

It is impossible to evaluate the camera calibration accuracy based on the accuracy of the estimated model parameters since the true model parameters are unknown. Considering that the 3D positions of test points can be obtained by the stereo-vision system, the 3D reconstruction accuracy is utilized to evaluate the calibration accuracy. The 3D reconstruction RMS error of calibration test points is calculated as the root-mean-square difference between the reconstructed marker 3D location from camera parameters and the CMM readings.

First, the effect of the lens distortion model on calibration accuracy is investigated. The major difference in the camera model is the lens distortion model. Three cases are compared here. Case 1: only the curvature distortion is considered. Case 2: curvature distortion and

decentering distortion are included. Case 3: all distortion effects in Eq (7) are included. Table I compares the 3D error residual of calibration test points in 3 cases. It is clear that the curvature distortion is the dominant component in the selected lens, which is reasonable because wide-angle lenses generally have more curvature distortion. In addition, incorporating the decentering distortion further improves the calibration accuracy. On the other hand, the effect of thin prism distortion is negligible in the selected lens.

Table I: Comparison of 3D calibration residual with different lens distortion models

case	1	2	3
Residual [μm]	20.8	14.8	14.3

The calibration accuracy is evaluated by using another set of 400 test points different from that of calibration. Based on the estimated model parameters, the 3D position reconstruction algorithms are implemented in a real-time computer. Since the calibration results show that the effect of thin prism distortion is negligible in the selected lens, the real-time distortion correction algorithm does not consider the thin prism distortion. Similar to the calibration procedure, an active LED marker is carried by the CMM in a grid of reference points filling the measurement volume. At each reference position, the 3D reconstruction result based on the estimated model parameters is compared with the result given by the CMM, and the 3D reconstruction error is recorded. A 3D measurement accuracy of $19.5\mu\text{m}$ is achieved after camera calibration. Figure 3 shows the error histogram in each direction. It is clear that the error distributions clearly follow a normal distribution with zero mean error. Finally, the 6-axis position sensor has been integrated into a magnetically-levitated 6-axis planar motion machine to experimentally demonstrate its effectiveness.

CONCLUSION

This paper reviewed the prior art of camera calibration methods and presented a high-accuracy camera calibration method using a CMM and active LED marker. Experimental results have demonstrated that a 3D position measurement accuracy of $19.5\mu\text{m}$ is achieved over a measurement range of 400mm by 400mm by 15mm after the camera calibration.

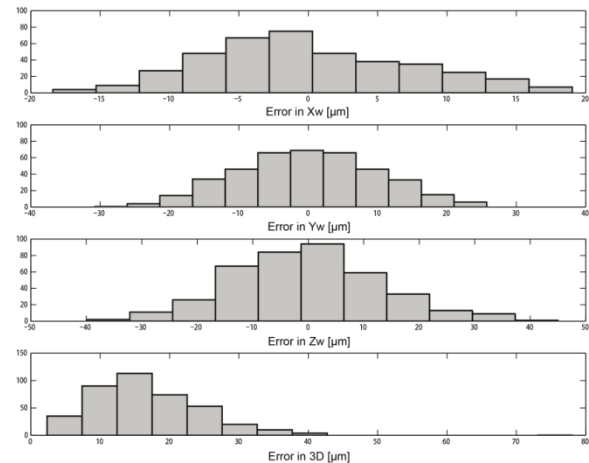


Figure 3: Histogram of 3D reconstruction error

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TRINANO N100 3D MEASUREMENTS WITH NANOMETER REPEATABILITY AND EFFECTS OF PROBE-SURFACE INTERACTION

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INTRODUCTION

In precision engineering, complex 3D components are developed and manufactured that are getting smaller and require increasingly tighter tolerances. As the components are smaller, their weight and the required measurement volume also decrease. In line with this trend, a novel coordinate measuring machine (CMM), the TriNano N100, has been developed, as shown in figures 1 and 2. The TriNano N100 design allows a 3D measurement uncertainty of 100 nanometers in a measurement range of 200 cubic centimeters, while greatly reducing manufacturing costs.

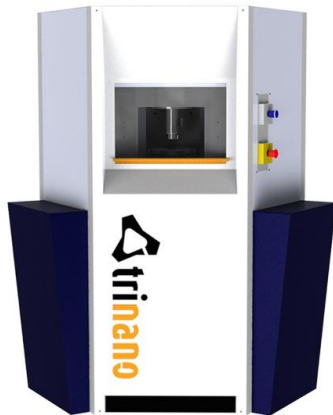


FIGURE 1. TriNano N100 CMM (artist impression).

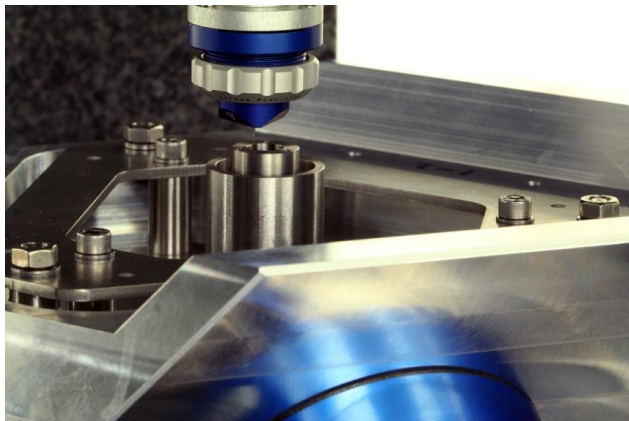


FIGURE 2. Photo of TriNano N100 measuring an artifact using a Gannan XM probing system.

This paper will discuss several important design aspects of this novel CMM as well as the results of the initial evaluation measurements that have been obtained.

PARALLEL DRIVE CONCEPT

In the TriNano, the workpiece is translated in 3 directions with respect to a stationary sensor, e.g. a tactile 3D probing system. This is achieved by three identical linear translation stages that support the workpiece table in a parallel configuration.

Each of the 3 stages performs a linear motion only, whereby the position of each stage is measured using a linear encoder. The measurement axes of all 3 linear encoders remain aligned to the center of the probe tip over the whole measurement volume, i.e. the CMM complies with the Abbe principle in 3 dimensions throughout its measurement range. This is schematically shown in figure 3 for 2 dimensions.

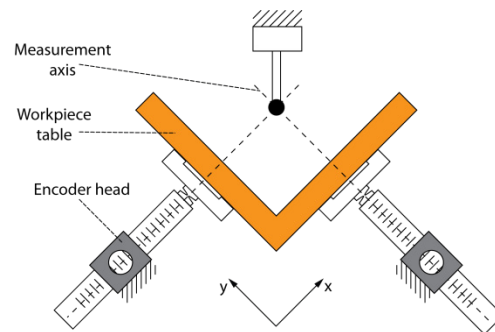


FIGURE 3. Schematic working principle 2D.

The stages are positioned orthogonally and in parallel and support the work piece table via vacuum preloaded (VPL) air bearings, as shown in figure 4. A linear translation of a stage is transferred via its VPL air bearing to the work piece table. Translations of the work piece table perpendicular to the translation direction of the linear stage are decoupled by its VPL air bearing. Thus, each stage controls the position

of the workpiece table in one direction only, i.e. the stages are decoupled. As a result of the relatively simple operating principle, consisting of 3 identical stages and the use of linear encoders, this CMM is relatively cheap to manufacture and maintain.

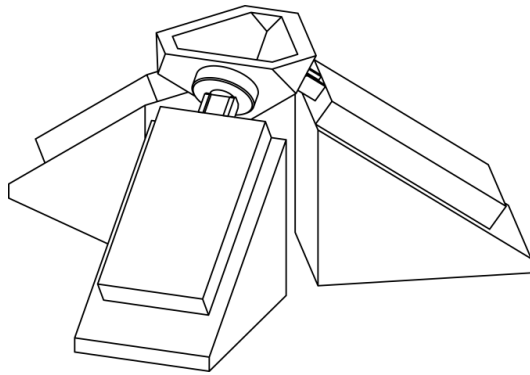


FIGURE 4. Schematic working principle 3D.

The parallel configuration of the three identical stages supporting the workpiece stages results in superior dynamic behaviour of the TriNano. This configuration allows a low and equal actuated moving mass of each stage with short and stiff structural loops. On machine measurements show that the lowest natural frequency in the positioning loop is 75 Hz. This allows a high control bandwidth, required for scanning measurements of micro parts with a velocity of 1-2 mm/s, and reduces the influence of (external) vibrations on the measurement.

THERMAL EFFECTS

Thermally induced errors are often the largest contribution to the total error budget in precision measurement equipment despite the research efforts performed on the matter [1,2,3]. In the TriNano, thermal effects have been reduced by using low expansion materials for critical parts of the metrology loop. Also, the effect of heat sources is reduced. One example is in the heat generation of the actuators. Dynamic forces are minimized by the parallel layout of the machine, whereby each actuator only accelerates the mass of one stage and the work piece table. Static forces are minimized by the use of a frictionless pneumatic weight compensation mechanism. By adjusting the pressure, this mechanism can be adjusted to compensate for the mass of different work pieces as well.

The workpiece table and sensor support in the metrology loop are expensive to manufacture

from low expansion materials like Zerodur® or Invar®. Therefore, these components are made from aluminium and compensation for their thermal expansion is used. NTC's are used to measure temperature variations in critical parts of the metrology loop. A linear compensation model is used to obtain an estimate for the thermal expansion. The main advantage of NTC's compared to other sensors like PT100's is their resolution [4] which is better than 0.1 mK.

MEASUREMENT RESULTS

All measurements as discussed in this section are performed using a TriNano N100 CMM with a Gann XP 3D probing system. The results include the disturbances of all parts of the metrology loop, e.g. thermally induced errors and the stability of the vacuum preloaded air bearings. The stability in the air gap of the vacuum preloaded air bearings is discussed in a previous publication [5].

Single point repeatability

The single point repeatability is determined by repeatedly measuring the same point on a steel gauge block. Each position is recorded and the deviation between the measured positions is calculated, as shown in figure 5.

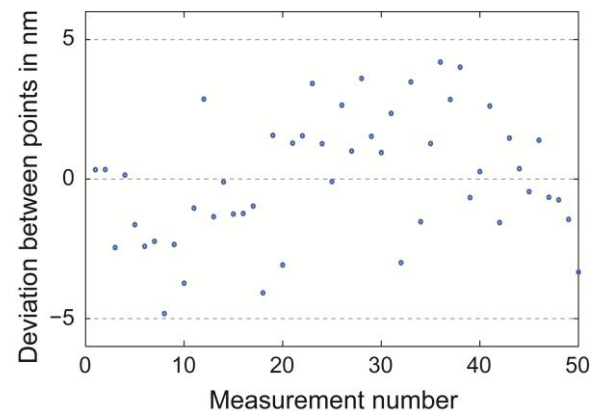


FIGURE 5. Result of a single point repeatability measurement in vertical direction. The standard deviation over 50 points in this measurement was 2.4 nm.

The standard deviation in the single point repeatability measurement shown in figure 5 is calculated to be 2.4 nm. This measurement was repeated at different position in the measurement volume, with similar results.

Repeatability in distance

To determine the repeatability in the distance measurement between points, two gauge blocks were placed in the CMM, as shown in figure 6. The result of the distance measurement between 2 positions on the gauge blocks is shown in figure 7. From this graph, the standard deviation in the measurement of the distance between 2 points is calculated to be 4.1 nm.

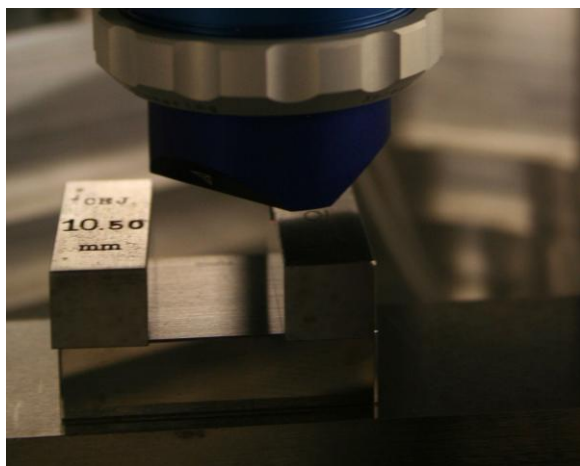


FIGURE 6. Photo of the repeatability measurement of the distance between 2 gauge blocks.

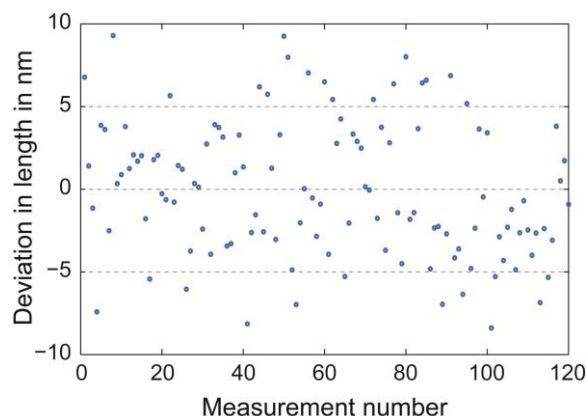


FIGURE 7. Repeatability in the measurement of the distance between 2 gauge blocks at a nominal separation of 13.876 mm. The standard deviation over 120 points in this measurement was 4.1 nm.

Long term stability and temperature compensation

Figure 8 shows the expansion in the metrology loop of the TriNano. During a 3 hour measurement a drift of 160 nm was observed. First results show that, after compensation for

thermal expansion, a top-top deviation of 28 nm is obtained, as shown in figure 9. As thermal shielding of the frame was not yet available, further improvements can be expected.

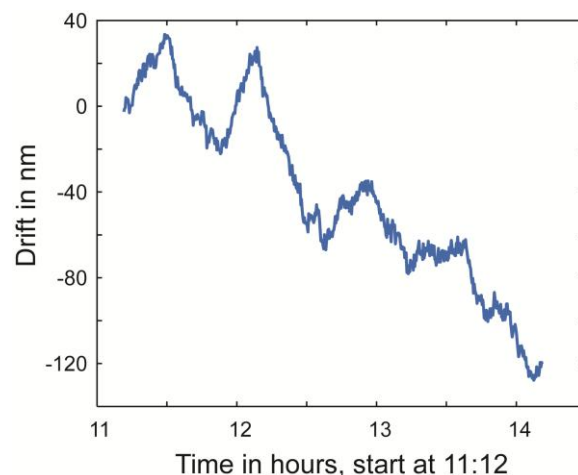


FIGURE 8. Drift in the metrology loop over a 3 hour measurement.

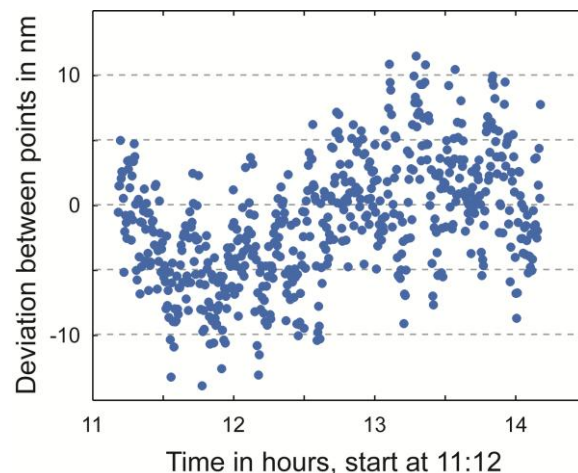


FIGURE 9. Single point repeatability over a 3 hour measurement after thermal compensation is applied.

Repeatability of a scanning measurement

To verify the dynamic behavior of all components in the metrology loop, including the Gannen XP probe, scanning tests are performed at a scanning velocity of 1 mm/s. The measurement object is an optical flat which is scanned using a Gannen XP probe with a ruby tip of 0.3 mm in diameter. The top surface of the optical flat is measured using two square patterns, as shown in figure 10. From this graph it can be seen that the optical flat was slightly tilted during the measurement.

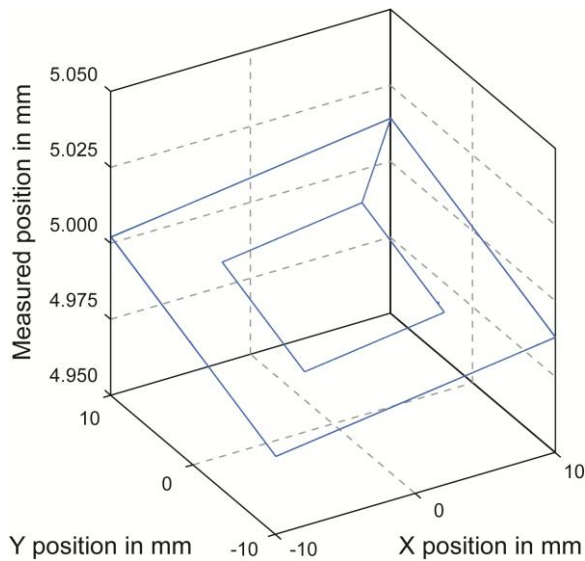


FIGURE 10. Scanned pattern, consisting of 2 squares, on the top surface of an optical flat.

The same pattern on the optical flat was measured twice. The difference between all corresponding measurement values of both scanning cycles, without averaging or filtering of data points, is within a band of ± 20 nm, as shown in figure 11.

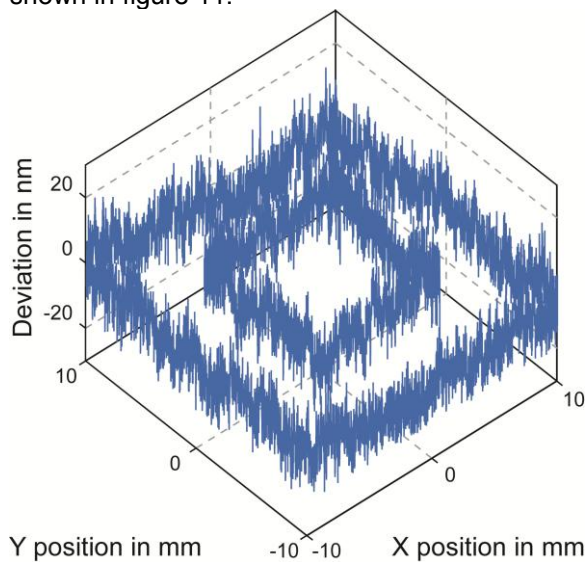


FIGURE 11. Deviations between the measured positions of individual data points for two scanning measurements on the top surface of an optical flat.

CONCLUSION

A novel CMM has been developed and manufactured. Its design allows nanometer

uncertainty, while greatly reducing manufacturing costs.

Initial verification measurements show a standard deviation in single point repeatability of 2.4 nm. The standard deviation in the distance measurement between 2 gauge blocks is measured to be 4.1 nm.

Furthermore, a thermal compensation algorithm has been implemented that allows long-term measurements to be performed with nanometer stability. Using this algorithm, a top-top deviation of 28 nm over a 3 hour measurement is obtained.

Finally, the CMM and probe dynamics have been investigated. The differences between repeated scanning cycles at 1 mm/s are shown to be within a band of ± 20 nm.

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PRECISION MASS PROPERTY MEASUREMENTS USING A FIVE-WIRE TORSION PENDULUM

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INTRODUCTION

A method for measuring the moment of inertia of an object using a five-wire torsion pendulum design is described here. Typical moment of inertia measurement devices are capable of 1 part in 10^3 accuracy and current state of the art techniques have capabilities of about one part in 10^4 [1]. The five-wire apparatus design, Figure 1, shows the prospect of improving on current state of the art. Current measurements using a laboratory prototype indicate a moment of inertia measurement precision better than a part in 10^4 . In addition, the apparatus is shown to be capable of measuring the mass center offset from the geometric center. Typical mass center measurement devices exhibit a measurement precision up to approximately $1\text{ }\mu\text{m}$. Although the five-wire pendulum was not originally designed for mass center measurements, preliminary results indicate an apparatus with a similar design may have the potential of achieving state of the art precision.

MEASUREMENT APPARATUS

An apparatus designed to generate a rotation about an axis provides the ability to measure an objects mass center and moment of inertia. The measurement apparatus typically attempts to produce a pure rotation about a single degree of freedom. When designing an apparatus to measure the moment of inertia to a high precision, care must be taken to minimize the extra degrees of freedom in the system. The measurements of rotation will have uncertainties when there are significant other degrees of freedom. Bifilar and trifilar pendulums do not constrain the swinging or lateral translation modes. To improve the accuracy of a standard trifilar pendulum, the lateral pendulum modes need to be constrained. Five support wires are sufficient to constrain all but one degree of freedom. In a five-wire pendulum, two additional wires are arranged as shown in Figure 1 to minimize rotations about the other two rotational axes. The design reduces errors due to tilt and horizontal translational degrees of freedom. The three attach points on the platform supporting the inertia to be measured are posi-

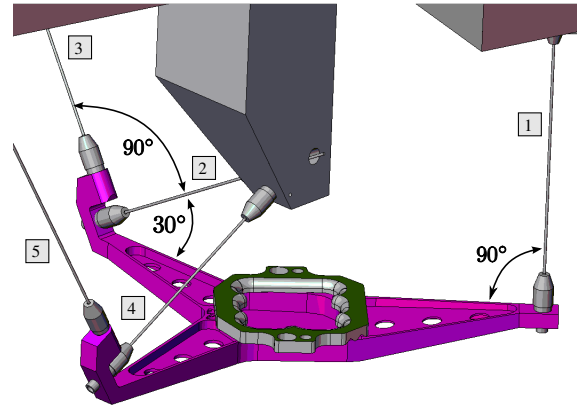


FIGURE 1. Five-wire torsion pendulum.

tioned equidistant from the center of rotation. At one attach point, a single vertical wire is used and the other two attach points consist of two wires. The horizontal components of the wires which are splayed out from a single attach point provide horizontal stiffness.

Moment of Inertia Measurement

In order to measure the moment of inertia, I , the object is rotated about an axis and the pendulum natural frequency, ω , is used to determine the radius of gyration, R_g , about that rotational axis. The relationship between the torsion pendulum rotational frequency and the radius of gyration is given by: $R_g^2 = I/m = k/\omega^2$, where k is the torsion coefficient or stiffness constant of the pendulum and m is the total mass. The full moment of inertia tensor contains six independent terms. As such, a moment of inertia measurement device must be capable of determining the radius of gyration about at least six different axes of rotation.

Mass Center Measurement

By measuring the pendulum natural frequency of rotation, the mass center offset from the geometric center of an object is also obtained. The instantaneous moment of inertia consists of the moment of inertia about the objects mass center plus the parallel axis theorem components of the objects mass times the square of the distance

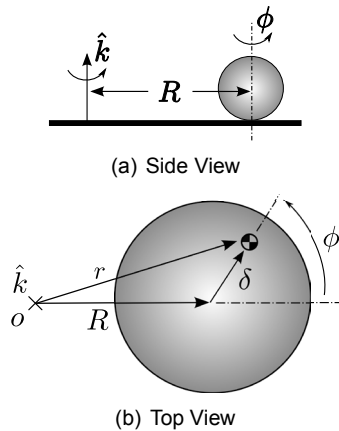


FIGURE 2. Mass center measurement configuration using a torsion pendulum.

from the rotation center to the mass center, r^2 . Placing an object on the pendulum far from the rotation axis amplifies the contribution due to the mass center offset from the geometric center, due to the quadratic dependence on r . By changing the orientation of the object with a fixed geometric location relative to the pendulum rotation center, the mass center offset is determined by measuring the change in the natural frequency.

Consider for example an object with a mass center offset from the geometric center of magnitude δ . The object's geometric center is placed on the pendulum at a location of R from the pendulum rotation axis, $\hat{k}$. Refer to Figure 2 for a graphical depiction. The mass center offset within a plane can be determined by rotating the object about an axis parallel to the pendulum rotation axis. The offset is determined by measuring the pendulum oscillation frequency for different rotation angles of ϕ . For each rotation angle ϕ , a corresponding change in the pendulum oscillation frequency will be observed.

MEASUREMENT OBJECT

Two different types of test objects are used to demonstrate mass property measurements with the five-wire pendulum. One object is specifically designed for evaluating moment of inertia measurements, while the other is intended for demonstrating mass center measurements.

For moment of inertia measurements, a preferred principal axis of inertia sphere is used as the test object. The spherical objects are designed such that the principal moment of inertia difference ratio, $(I_{p33} - I_{p22})/I_{p11}$, is greater than 10 %. A

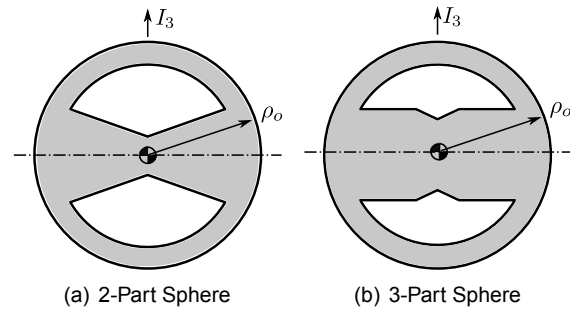


FIGURE 3. Preferred principal axis of inertia sphere simplified cross-section. The cross section view for the 2-part sphere (a) is in the plane of the seam between the two hemispherical shells. The cross section view for the 3-part sphere (b) is in a plane perpendicular to the plane of the two seams for each hemispherical shell.

set of spheres were fabricated as part of a student project in the Precision Engineering course taught at Stanford University by Professors Dan DeBra and Dave Beach [2]. The preferred principal axis of inertia spheres are constructed of either two or three brass parts and then brazed together to create a complete sphere. The moment of inertia difference ratio for the spherical geometry is designed to be on the order of 10 %. Figure 3 shows a simplified cross-sectional view of the two designs.

For mass center measurements, a sphere with a fixed mass center offset is used as the test object [3]. A 2 inch (50.8 mm) diameter BeCu sphere was drilled with two holes forming a "V" to produce a mass center offset with respect to the geometric center. The sphere is designed to have a theoretical mass center offset of approximately $\delta = 40 \mu\text{m}$.

Prior to the measurement process, the spheres must be marked to define a reference frame. Two perpendicular great circles are drawn onto the surface of the spheres to be used as a coordinate reference. On each of the fabricated preferred principal axis of inertia spheres, the seam is faintly visible as a silver line resulting from the brazing material. One great circle is drawn in a plane parallel to the visible brazed seam. The other two measurement planes are located mutually perpendicular to the first great circle. For the fixed mass center offset sphere, mutually perpendicular great circles are also drawn on the surface. The first great circle is drawn to intersect the two drilled holes comprising the "V". Again, the

other two measurement planes are located mutually perpendicular to the first great circle.

MEASUREMENT RESULTS

The torsion pendulum is designed to rotate about a single axis. As such the measurement object, i.e. the sphere, must be rotated to different orientations with respect to the rotation axis. The choice of these orientations must adequately map out the inertia ellipsoid. For the preferred principal axis of inertia spheres, a set of nine measurement configurations was performed, with rotation angles at $\pi/4$ radian increments. For the fixed mass center offset sphere, measurements are also performed at $\pi/4$ radian increments, for eight measurements in each plane. Measurement configurations are chosen to adequately capture the minimum and maximum location of the mass center relative to the pendulum rotation axis. At the time of this publication, mass center measurements have been conducted in two of the three independent planes and the data analysis has been completed on the first measurement plane.

Moment of Inertia

In a five-wire torsion pendulum shown in Figure 1, five wires are arranged to minimize translational degrees of freedom, thereby reducing errors due to tilt and translation. The desired measurement parameter is the pendulum natural frequency of oscillation. The oscillation frequency is extracted from the recorded pendulum position time response data. By reducing the errors associated with generating a pure rotation, the five-wire pendulum has demonstrated a consistent frequency measurement to better than a part in 10^5 . The oscillation frequencies for the the five-wire pendulum are on the order of 2.5 Hz to 4.0 Hz for the unloaded and loaded pendulum configurations respectively.

For the preferred principal axis of inertia sphere measurements, the frequency of oscillation was measured from about one part in 10^5 to a few parts in 10^6 for the various measurement orientations. Table 1 shows the resulting moment of inertia tensor measurement values for the preferred principal axis of inertia spheres. The results indicate a repeatable moment of inertia measurement on the order of a part in 10^4 or better.

For the moment of inertia measurements, the five-wire pendulum was calibrated with the two-part sphere on the platform. The calibration procedure was not repeated for the 3-part sphere.

As a result, the indicated measurement results for the 3-part sphere may contain additional systematic errors as compared to the 2-part sphere measurement results. A different loading of the pendulum platform may result in subtle changes to the pendulum stiffness coefficient, which is estimated by the calibration process. Still, the repeatability of the measurements is the true indication of the precision associated with the measurement apparatus.

Mass Center

Although the constructed five-wire torsion pendulum was not originally designed to perform mass center measurements, the torsion pendulum has demonstrated precision measurement results. As already indicated, the measurement object for the mass center measurements is a specially designed fixed mass center offset spherical mass. By assuming a homogeneous density and an ideal geometry of a perfect sphere with the removed material, one can estimate a theoretical mass center offset value. Furthermore, by measuring the drilled hole locations and geometry of the actual fabricated sphere, the expected mass center location can be further estimated. Yet, this object has been used previously in other mass center measurement experiments and therefore has historical measurement results for comparison. As a reference, this work uses the values obtained by Conklin [4]. Although this particular sphere has only had the mass center offset measured to approximately $1\text{ }\mu\text{m}$, Conklin has shown that the applied velocity modulation measurement technique can be used to measure the mass center location of spheres to state of art precision on the order of $\sim 0.1\text{ }\mu\text{m}$.

The five-wire pendulum has been used previously to demonstrate the feasibility of mass center measurements with the torsion pendulum apparatus. These previous efforts estimate the mass center offset using a two configuration measurement approach. These two configurations correspond to the minimum and maximum radial distances from the pendulum rotation axis. Clearly, the previous measurements are approximate and the result is expected to be an under estimate of the true value.

This work further investigates the use of the five-wire pendulum for mass center measurements, which utilizes a more complete set of measurement configurations. As already indicated, eight measurements are performed in each measure-

Sphere	m_o kg	$[I_p]$ kg·mm ²	$[\Sigma_p]$ kg·mm ² × 10 ⁻²	σ_{p_i}/I_{p_i} × 10 ⁻⁴	$\frac{I_{p33}-I_{p22}}{I_{p11}}$
2-Part	0.4489	$\begin{bmatrix} 112.67 & & \\ & 112.98 & \\ & & 126.41 \end{bmatrix}$	$\begin{bmatrix} 1.1 & & \\ & 0.2 & \\ & & 0.1 \end{bmatrix}$	$\begin{bmatrix} 0.94 & & \\ & 0.19 & \\ & & 0.05 \end{bmatrix}$	0.1192
3-Part	0.4551	$\begin{bmatrix} 113.15 & & \\ & 113.60 & \\ & & 126.50 \end{bmatrix}$	$\begin{bmatrix} 1.7 & & \\ & 0.1 & \\ & & 1.1 \end{bmatrix}$	$\begin{bmatrix} 1.50 & & \\ & 0.12 & \\ & & 0.89 \end{bmatrix}$	0.1141

TABLE 1. Measurement results for preferred principal axis of inertia spheres.

ment plane. The pendulum frequency of oscillation was measured to a couple parts in 10⁵ or better for the various measurement orientations. Using the eight equally-spaced measurements for the measurement plane with the largest mass center offset component, the mass center offset was determined to be a magnitude of 41.89 μ m within the rotation plane. Repeatable frequency measurements with the five-wire pendulum result in an estimated mass center offset measurement error of better than 0.8 μ m. It should be noted that a full error analysis on the mass center measurements has not yet been completed. The mass center measurement results contain for example systematic errors associated with the measurement apparatus calibration procedure. The measurement apparatus calibration parameters utilized for the mass center measurements are those used for the moment of inertia measurements. A slight change in the calibration coefficients and the measurement results correspondingly is expected. Table 2 summarizes the current and historical measurements for the mass center offset from the geometric center for the fixed mass center offset sphere.

SUMMARY

The five-wire pendulum is capable of determining moment of inertia properties to better than a part in 10⁴. The prototype five-wire measurement apparatus shows the prospect of further improving on current state of the art precision. The apparatus is also capable of mass center offset measurements to better than ~0.8 μ m. Further work is necessary to completely characterize the error associated with the mass center measurements. For mass center measurements, an apparatus with a similar five-wire design shows the prospect of achieving state of the art. For the mo-

Source	MC Offset	Notes
(a)	40.0 μ m	Theoretical, [3]
(b)	44.0 μ m	$\sigma = 10$ μ m, [4]
(c)	41.17 μ m	$\sigma = 1.5$ μ m, [4]
(d)	40.23 μ m	Error: ~1 μ m, [5]
(e)	41.89 μ m	Error: ~0.8 μ m

TABLE 2. Comparison of measurement results for the fixed mass center offset sphere. Measurement sources: (a) Design value, (b) Estimate from geometry, (c) Velocity modulation, (measured), (d) Torsion pendulum, (Single Plane, measured, two-configuration estimate). (e) Torsion pendulum, (Single Plane, measured).

ment of inertia measurements, current limitations are associated with environmental disturbances, primarily vibrations induced by human motion. A full description of the associated pendulum measurement errors is available in [5].

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ERROR BUDGETING AS APPLIED TO THE DESIGN OF THE INTERNATIONAL X-RAY OBSERVATORY MANDREL MEASURING MACHINE

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This abstract discusses error budgeting as applied to the metrology for production control of a large number of ultra-precision X-ray telescope forming mandrels. In addition, this discussion outlines the manufacturing challenge, a proposed metrology solution, and an error budget demonstrating the feasibility of a conceptual instrument design. (July 30, 2012)

BACKGROUND

The International X-Ray Observatory, scheduled to be launched in 2021, is to be a 15,000 element Wolter Type I telescope sponsored jointly by NASA, the European Space Agency, and the Japan Aerospace Exploration Agency.

The subject telescope is to be assembled from concentric rings, composed of a total of approximately 15,000 elements, as illustrated in Figure 1.

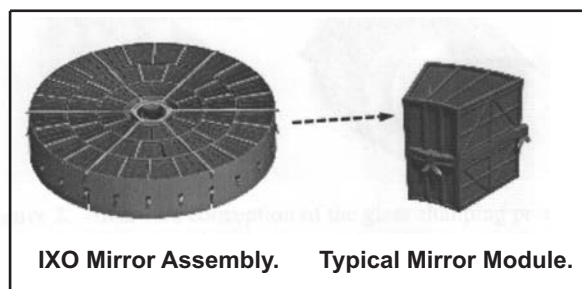


Figure 1. IXO flight mirror assembly and how it is divided into modules.

Individual components of the telescope are projected to be manufactured by ‘slumping’ glass over fused quartz or stainless steel mandrels, of which there will be approximately 700.

The mandrels are to be manufactured using computer-controlled polishing, which, while deterministic, will require process-intermittent or in-process metrology to monitor and control convergence of the process.

REQUIREMENTS

Typical tolerances on individual elements are of the order of 2.0 nanometers rms and 0.4 arc-seconds, as outlined in Table 1.

Mandrel Surface Parameter		Error Allocation for either Primary or Secondary (assuming perfection for the other)		RSS Contribution to HPD (arcsec)	Cumulative HPD (arcsec)
		Unit	Numerical Value		
Radius	Average Radius	μm	10.00	0.2	0.2
	Radius Variation	μm	0.20	0.1	0.2
Cone Angle	Average Cone Angle	arcsec	0.50	0.7	0.8
	Cone Angle Variation	arcsec	0.40	0.7	1.0
Axial Sag	Average Sag	μm	0.05	0.5	1.1
	Sag Variation	μm	0.05	0.2	1.1
Axial Figure	Low Frequency Figure (200mm-20mm)	nm	5.00	0.4	1.2
	Mid Frequency Figure (20mm-2mm)	nm	2.00	0.9	1.5
	High Frequency Figure (2mm-0.002mm)	nm	NA	NA	1.5

Table 1. Error Budget for Individual Forming Mandrels.

The largest components have a nominal radius of over 3.2 m, an axial length of 200 mm, and a chord width of 700 mm. The thickness of each element is less than 1.0 mm.

The shape and aspect ratios of individual elements means that interferometric metrology is not a practical option for inspection.

SOLUTION

Under a NASA-sponsored contract, the authors have designed a unique point-defined measuring machine for the process-intermittent metrology of individual mandrels. See Figures 2 and 3.

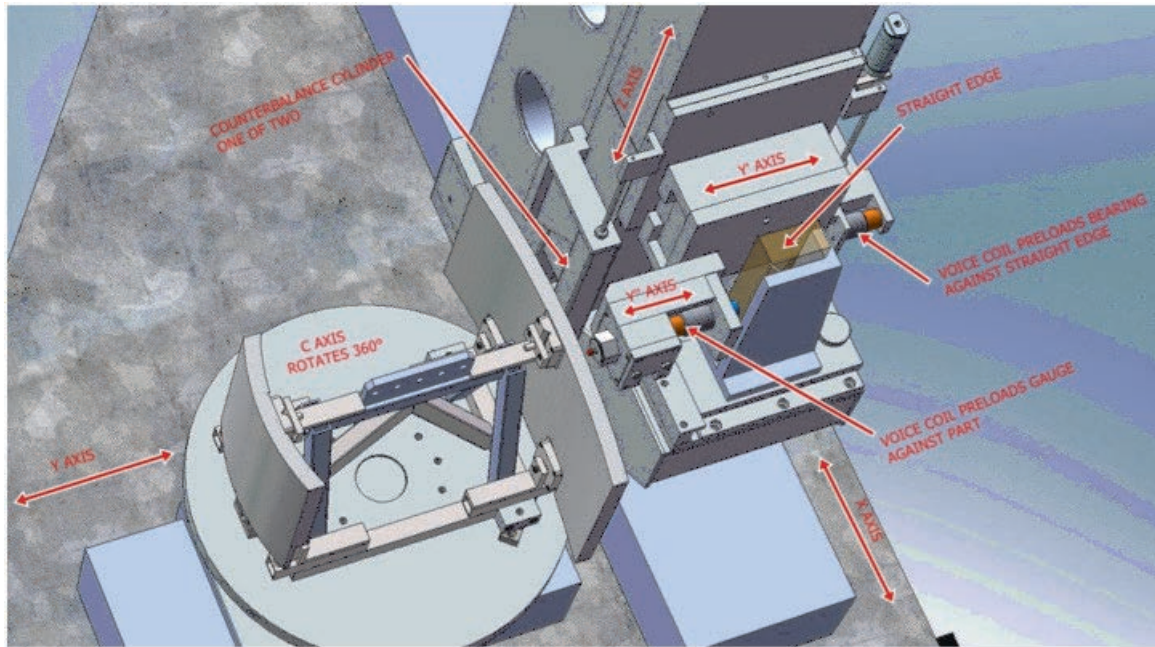


Figure 3. The six axes of the IXOMMM.

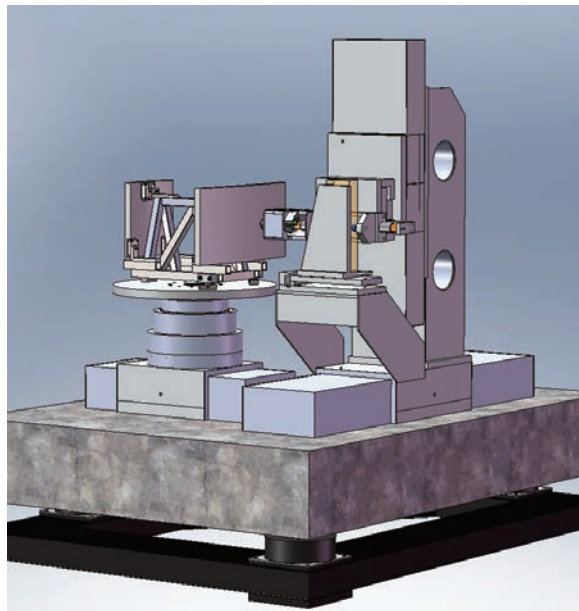


Figure 2. Preliminary Design of IXOMMM.

The authors designed the stand-alone instrument to capture the most critical component errors, for feedback to the manufacturing process and final accuracy confirmation. The design features both traditional and new design elements: careful attention to frictionless motion (e.g., air-bearings, linear motors, linear scale feedback), the Abbe Principle, extensive use of low CTE materials, temperature control (to $\pm 0.005^\circ\text{C}$), the application of a “virtual axis of rotation” to minimize the structural loop, and the incorporation of a unique ultra-low force contact probe.

A complete description (with illustrations and axis nomenclature) of the proposed IXOMMM follows. Key features of the instrument include:

- Design of the instrument to use a “virtual axis of rotation.” While individual mandrels are axisymmetrical segments, their radii of curvature make manufacturing and metrology on-axis impractical. Thus the proposed instrument is designed to measure the components off-axis, while simultaneously preserving axis, radius, radius deviation and all other specified information, and minimizing the structural loop of the machine.
- The use of an ultra-precision reference straight-edge for axial measurements. Because the tightest tolerances on individual mandrels are specified in the axial direction, a Zerodur straight-edge (flatness of 5 nm rms overall in two dimensions—2 nm in straightness over critical dimensions) will be used to guide the probe (using an air-bearing ‘follower’) during axial measurements. The absolute straightness accuracy of axial measurements will be determined by (1) the effective straightness of the straightedge, (2) the averaging effect introduced by the use of an air-bearing follower, (3) non-repeatability introduced by thermal and other effects, and (4) if necessary, the use of a second reference straightedge, and the application of reversal techniques to eliminate systematic errors.
- Ultra-precision probing technologies. The angles of inclination of the mandrel surfaces to the axis of rotation dictate that the probing system have a travel of

approximately 5 mm. Maintaining the probe system positioning accuracy to 1 nm rms suggests that the combined uncertainties of the probe-position location measuring system and the probe itself be less than 1 part in 10^6 . Several solutions present themselves for probe positional feedback (Y" axis), ranging from a laser interferometer operating in vacuum (expensive and complex), to an ultra-precision linear scale. For the purposes of the current design, we have selected a Magnascale laser scale. The probe itself is designed to be a contact device, with probing force closed-loop controlled to 1.0 mN through the use of a voice coil. The active portion of the probe will be guided by air-bearings in order to eliminate force variation and hysteresis. Significant experience with ultra-low force gauging using low friction styli (e.g., ruby, sapphire or diamond) have proven to provide (1) zero detectable part damage, (2) immunity to dust, fluids or other contaminants on the surface of the part, and (3) immunity to wear. Substantiating information by Thompson et al is provided in (Ref. 4). The overall dynamic range of the Y" probe axis is 30 mm.

- Position feedback—X, Y, and Z axes. The requirements for other motion axes are reasonably tight, but considerably more relaxed than those normal to the part surface. Thus it is anticipated that commercial ultra-precision scales can be used to provide feedback (both for feedback and data acquisition) in the C, X, Y, and Z directions.
- Temperature control. The extremely tight axial metrology specifications will require the application of state-of-the-art temperature control in order to maintain dimensional stability; extensive finite element modeling will be required to specify exact requirements, although experience suggests that maintaining the thermal environment to within 0.01 °C p-v will be required. In addition, the use of low CTE materials (e.g., Zerodur) will be required to maintain the dimensional stability of critical components with relatively short thermal time constants (e.g., the probe assembly, reference straightedge).
- Electro-mechanical features. The IXOMMM will be designed to be a frictionless instrument. All drives (both linear and rotary) will be brushless DC. All bearings (including those in the gauge head) will be hydrostatic air. Counterpoise cylinders for the Z axis will be air-bearing. Damping will be electrical. As mentioned above, all axes not being actively used in a measurement will be deactivated and locked, using kinematic, non-influencing mechanical clamps. Thus, for example, during an axial trace the X, Y, and C axes will all be locked; the Z axis will sweep the output of

the probe recorded for post-process analysis. The C axis will then be unclamped, indexed, re-clamped, and the process repeated until all axial traces are complete.

ERROR BUDGETING AS A DESIGN TOOL

The error budget has become an essential tool for the designer of precision machine tools and other precision systems. As defined by Donaldson (Ref. 1), "...an error budget is a systems analysis tool, used for prediction and control of the total error of a system at the design stage for systems where accuracy is an important measure of performance. Given a system-error goal, an error budget can be used in a control mode to set individual subsystem error limits, while also making trade-offs that balance the level of difficulty among the subsystems. In a predictive mode, proposed subsystem design can be assessed for error contributions, leading to a predicted overall system error."

Accurate input to an error budget can be derived from a number of sources. Typically the input is in the form of parametric error values for individual sources of error. Thus the use of proven subsystem design concepts allows the measurement of error values, and their direct insertion into the error budget. In those instances where empirical data is not available, a variety of analytical tools can be used. Most notable is the application of finite element modeling to assess both subsystem and overall system error sources.

The process of design using error budget analysis as a tool is obviously iterative. Individual sub-systems are evaluated and optimized for cost and performance, while simultaneously an error budget is used to predict overall system performance. The error budget allows the designer to assess, for example, the impact of a cost-motivated design change on final system performance. Continued iterations of the design, analysis, error budget sequence can be used to ensure that costs can be optimized without compromising performance goals.

Implicit in this process is the assumption that significantly exceeding performance goals are as undesirable (from a cost standpoint) as failing to meet performance goals. Thus, the accuracy of the error budget is critical to the success of the design process.

ERROR BUDGET—COMBINATORIAL RULES

Specific accuracy terms for the IXO mandrel error budget are specified in different terms. The most critical (axial figure and sag) are specified in rms terms, while radius and cone angle are specified as p-v. The selection of combinatorial rules for the various error

sources in error budgeting their consequences is non-trivial. In the case of the rms specifications, the process is fairly straightforward; although in this instance the small number of error sources complicates the process somewhat.

The selection of a combinatorial rule for p-v errors is more complex. A well-tried and proven rule, however, is described by Thompson in 1995 (Ref. 7):

The greatest difficulty in using error budgeting for predicting absolute machine performance stems from the lack of a mathematically defensible combinatorial rule for summing individual error source values into a single performance figure. Lacking sufficient information about the spatial or temporal distribution of individual error sources, an overall upper bound error for a given set of conditions may be obtained by arithmetically summing all of the individual error values for a given coordinate direction... This approach is obviously extremely conservative; the probability of all errors reaching maxima at the same point in time and space with the same directional sense is very unlikely. The risk of basing an error budget on this approach is one of making the machine unnecessarily expensive. Alternatively, if all the errors are assumed to be uncorrelated, the rms sums of the directional errors can be used to establish a lower bound overall error. In this case, however, the predicted error would be overly optimistic, since many of the errors cannot be assumed to be uncorrelated and some are dependent on common sources.

Thompson also states:

In (a) 1989 CIRP paper we proposed a combinatorial rule, originally suggested by Hocken, that predicted an overall system error value based on the means of the upper (arithmetic sum) and lower (rms) bounds. While admittedly arbitrary, the proposed methodology appeared to yield results reasonably consistent with actual accuracies measured on existing precision machine tools... (Ref. 7).

The above approach to predicting p-v instrument errors has proven to be remarkably robust. "Hocken's Rule" is now well accepted in the field of precision engineering.

ERROR BUDGET FOR THE CONCEPTUAL DESIGN

Table 2 is a summary of the final error budget summary for the IXOMMM conceptual design, indicating that all of the mandrel errors could be measured to the required tolerances. This was the case for cone angle,

axial sag and axial figure specifications, using error values of 2 ppm p-v straightness and positioning accuracy, and 1 arc-sec for angular motion errors over the effective travel of a given axis.

Initial predictions for average radius error of the largest components did not meet specification. While the specified error value was by far the largest at 10.0 μm p-v, the initial assumptions led to an error prediction of 53.0 μm p-v for the largest radius part. This was not predicted at the outset.

The methodology used for calculating the average radius error was to (1) generate the arithmetic sum of the individual errors in the X and Y directions, (2) generate the RMS sums of the individual errors, (3) calculate the mean of the two sums (Hocken's rule), and then (4) apply the formula:

$$r = h^2/8s + s/2$$

where r is the actual radius, s is the actual sagittal depth, and h is the actual chord length. The actual radius is then subtracted from the nominal value to establish the average radius error.

The out-of-spec predicted error value for average radius was addressed by assuming (1) the application of error correction to slide straightness to achieve 1 ppm accuracy (a proven capability), (2) the selection of linear scales to achieve 1 ppm positioning accuracy (a proven option), and (3) the application of error correction to achieve angular motion accuracies of 0.1 ppm (a proven capability). Applying the same methodology to the resulting figures, an average radius error value of 8.9 μm p-v was predicted, well within the specified 10.0 μm specified. Table 3 summarizes the final error budget for average radius.

CONCLUSIONS

In this exercise the value of the error budgeting process was demonstrated in several ways:

1. The design/error budgeting cycle is an iterative process—and it is just as likely that a first design will be overly conservative (and therefore expensive) as not.
2. Results can be non-intuitive—as exemplified in this case by the largest single error category being the most difficult to achieve.
3. Some errors increase exponentially with dimension. Examine all error sources, and their consequences through the full range of part dimensions.

Mandrel Surface Parameter		Error Allocation for Either Primary or Secondary (assuming perfection for the other)		Errors Predicted by Error Budget
		Unit	Numeric Value	
Radius	Average Radius	μm	10 p-v	8.9 p-v
	Radius Variation	μm	0.20 p-v	0.14 p-v
Cone Angle	Average Cone Angle	arc-sec	0.50 rms	0.28 rms
	Cone Angle Variation	arc-sec	0.4 rms	0.28 rms
Axial Sag	Average Sag	μm	0.05 rms	1.80 rms
	Sag Variation	μm	0.05 rms	1.80 rms
Axial Figure	Low Frequency Figure (200 mm - 20 mm)	nm	5 rms	1.80 rms
	Mid Frequency Figure (20 mm - 2 mm)	nm	2 rms	1.80 rms
	High Frequency Figure (2 mm - 0.002 mm)	nm	NA	NA

Table 2. Error Budget Showing That All IXO Metrology Tolerances Can Be Met.

Error Source	P-V Amplitude (Corrected)	Effective Travel	Y-Direction Error (s) (Corrected)	X-Direction Error (h) (Corrected)
Y-axis Positioning	1 ppm	92 mm	92 nm	0
X-axis Positioning	1 ppm	350 mm	0	350 nm
Z-axis Positioning	1 ppm	100 mm	N/A	N/A
X-Axis, Y-Direction Straightness	0.1 ppm	350 mm	35 nm	N/A
X-Axis Roll	0.1 arc-sec	350 mm	175 nm	N/A
Y-Axis Pitch	0.1 arc-sec	46 mm	23 nm	N/A
Arithmetic Sum			325 nm	525 nm
RMS Sum			191 nm	151 nm
Predicted Error			258 nm	338 nm

$$r_{error} = 8.9 \mu\text{m} (<\text{specified } 10 \mu\text{m})$$

Table 3. Average Radius Sensitive-Direction Errors with Correction Applied.

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PLANAR MAGNETIC 6D LEVITATING SYSTEM WITH COMPACT SENSOR

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INTRODUCTION

The paper describes the structure and some selected model based design steps of a novel 6 DOF planar magnetic drive system with nanometer positioning accuracy. The movable platform of the system is fully passive and therefore free of any cable connection. The upper surface of the platform is freely accessible.

6 DOF LEVITATION SYSTEM

In the first prototype system the slider-platform can be moved 100 mm x 100 mm in x-y and 100 μ m in z-direction. To increase the efficiency and to minimize the control effort the motor contains 3 Halbach arrays in the slider and 3 coil pairs on the stator.

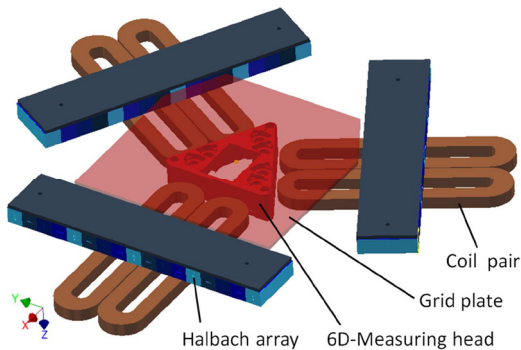


FIGURE 1. Schematic structure of the Prototype system

In addition the sliders lower side carries a centered grid plate that is scanned by the fixed measuring head placed on the stator in the middle of the coil pairs.

HALBACH ARRAY CONFIGURATION

Halbach array configurations are known for their high flux density on one side and small stray fields on the opposite side. FIGURE 2 shows a schematic of an array with a FEM result of the magnetic field. By commutating the coil pair depending on the slider position vertical and horizontal forces can be generated.

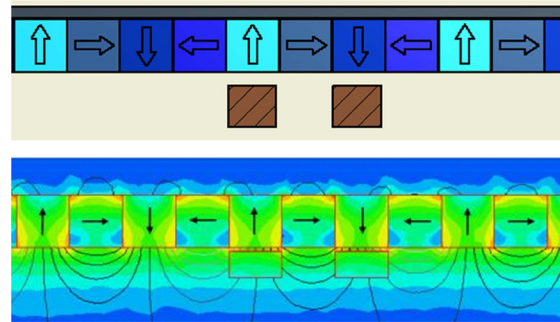


FIGURE 2. Model of a Halbach array with one coil and FEM result

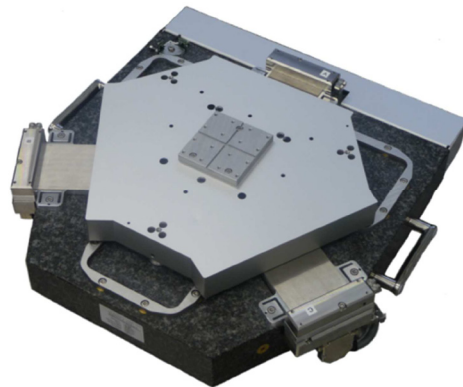


FIGURE 3. Experimental setup of the planar magnetic levitating system

The high field components below the magnets minimize the power consumption of the coils. Due to that characteristic they were chosen for the setup. Because of some strong repulsive forces during assembling of the Halbach arrays manufacturing tools were developed to align and glue the magnets precisely together. The chosen magnets were $\text{Sm}_2\text{Co}_{17}$ with medium magnetization grade resulting in flux densities of about 0.5 T. This material is also ultra-high vacuum compatible and can be used in applications that require a strong hydrogen atmosphere.

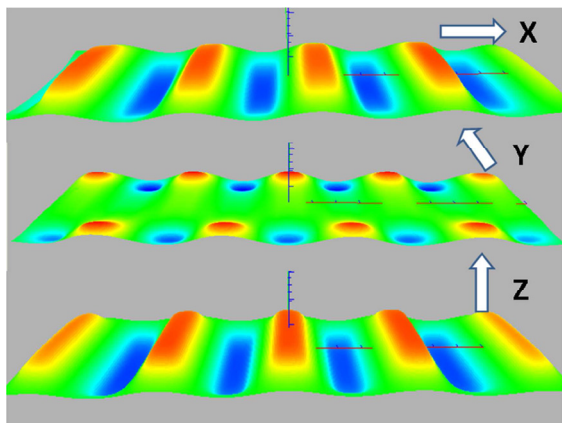


FIGURE 4. Measured flux components X, Y, Z for one Halbach array with 17 Elements

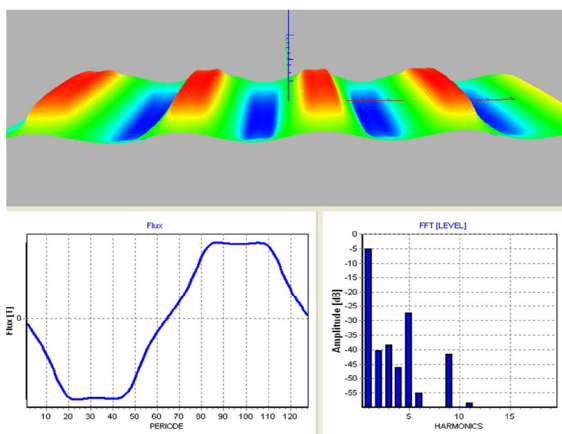


FIGURE 5. Measured horizontal flux component X, 1 mm above the Halbach array, harmonic distortion of 5th and 9th order

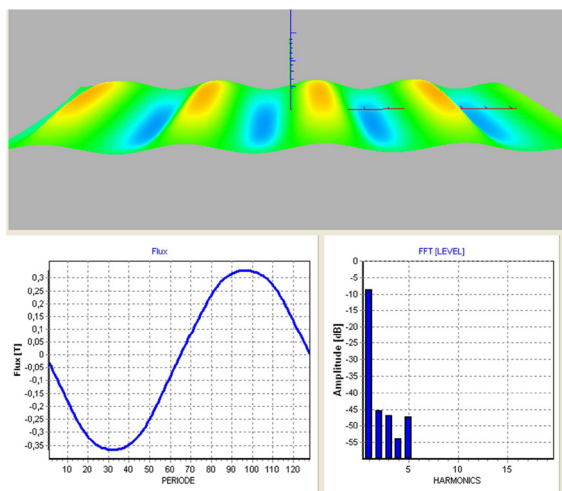


FIGURE 6. Measured horizontal flux component X, small distortions in 4 mm distance

Some measurement results of the flux density distribution at one Halbach array are shown in FIGURE 4 to FIGURE 6. The Y component of the flux density (see FIGURE 4) can be neglected since it runs parallel to coil current and therefore it doesn't produce any forces. Close to the Halbach array the X and Z fields show harmonics of 5th and 9th order (FIGURE 5). The coil geometry that is exposed to the magnetic field works like a filter function. Finally the forces are almost sinusoidal as shown in FIGURE 7.

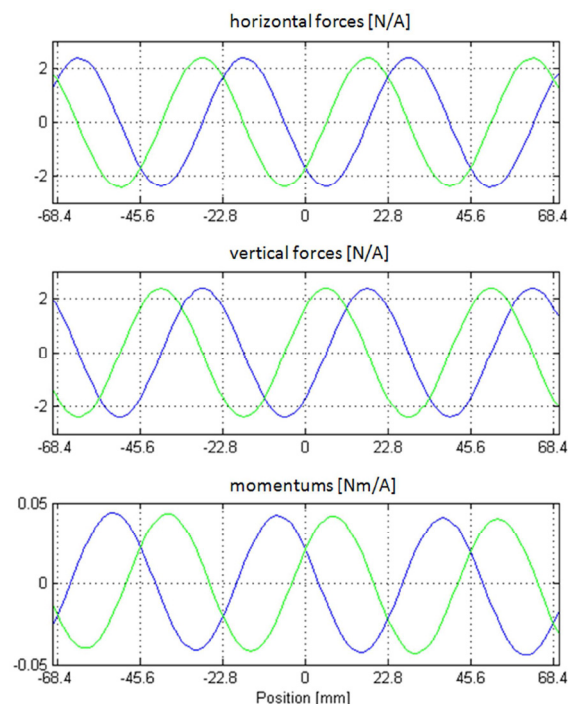


FIGURE 7. Calculated forces and momentums at a Halbach array, caused by the left (blue curves) and the right (green curves) coil strand according to FIGURE 2

COMMUTATION OF COIL PAIRS

The commutation of the 3 coil pairs is based on FEM results. The forces and momentums are given in a coil-fixed coordinate system and can be converted in a slider-fixed Cartesian coordinate system. Thus the position depending relation $K(x,y)$ between the 6 coil currents $i_1...i_6$ and the forces F_x , F_y , F_z as well as the momentums M_x , M_y , M_z is achieved.

$$\begin{pmatrix} F_x \\ F_y \\ F_z \\ M_x \\ M_y \\ M_z \end{pmatrix} = K(x, y) \cdot \begin{pmatrix} i_1 \\ i_2 \\ i_3 \\ i_4 \\ i_5 \\ i_6 \end{pmatrix} \quad (1)$$

The (6x6)-Matrix $K(x,y)$ is invertible in the whole travel range and thus forms the basis for the commutation of the coils. With the inverse Matrix $K(x,y)^{-1}$ the force set points, calculated by the controller, can be converted into the set points for coil currents.

The 36 elements of the 6x6 matrix K are calculated in the controller by a bicubic (spline) interpolation. In addition 5 correction matrices are used to compensate the differences between the Matrix $K(x,y)^{-1}$ calculated from FEM-results and the true relation between forces/momentums and coil currents.

THERMAL MANAGEMENT

Since the weight of the slider is continuously carried by the drive coils significant losses have to be handled in the coils. FIGURE 8 shows the losses in all coils to carry the slider weight including payload of together 5.3 kg depending on the x-y-position of the slider.

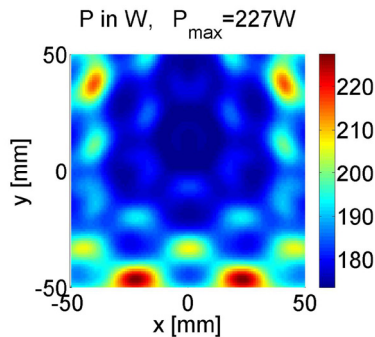


FIGURE 8. Sum of power losses in all coils to carry the load of 5.3 kg versus the slider position

Temperature stability is an important issue for highly accurate positioning technology. Carrying the load leads to a temperature rise on the top of the coil cooling sandwiches of approximately 1 K in centered position. Those small values are realized with cooling sandwiches based on efficient liquid pipe technology. Further improvements are under progress using lighter slider structures, more efficient magnetic circuits and better coil housings.

ROBUST SENSOR SYSTEM FOR 6 DOF

To get high stability in all six degrees of freedom a new 6D sensor system was developed. It consists of a 6D-optical-capacitive measuring system which uses a structured grid plate and allows some several hundreds of micro radiant for all rotational axes. The sensor resolution is in the nanometer and tenth of micro radiant range. A main advantage is the passive grid plate inside the movable platform. Therefore the sensor system allows the access to all sides of the platform, a requirement in inspection systems.

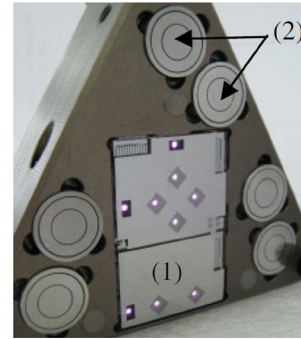


FIGURE 9. 6D-measuring head of the levitating system; (1) x-y-r_z-sensor; (2) cap-sensor pair

DIGITAL CONTROLLER

The control for all DOF of the platform is based on PI's digital industrial modular controller concept. The coils are driven by current amplifiers that are actuated by signals of the PI controller. The current set points for the amplifiers are calculated from the output of six servo loops, using the position dependent commutation matrix $K(x,y)^{-1}$, which allows a stabilized lifting and movement of the platform in all six DOF over the whole travel range.

The real time operating system of the CPU is synchronized by a FPGA interrupt mechanism. The controller runs with a PC-ETX processing module with a servo cycle time up to 200 μ s. Trajectory generation, simple to use command set and PI's real time interface is implemented. Coordinate transformation related to the sensor geometry is done in the controller. An initialization procedure is developed to drive the system in a well-defined starting position.

QUALIFICATION RESULTS

The prototype shows position stability in the resolution of the sensor system. Also for fast dynamic motion sequences the reaction forces are asymptotic limited.

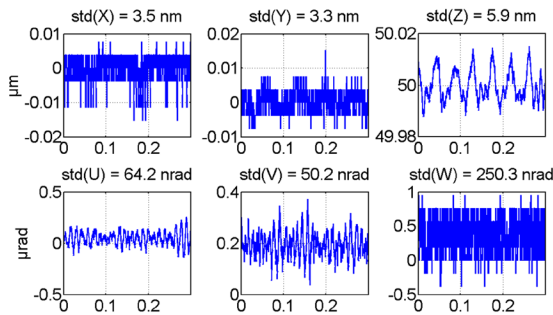


FIGURE 10. Position stability at position $x=y=0$, $z=50\mu\text{m}$; $u=v=w=0$

Horizontal movements with 50 mm/s and 2 m/s^2 are done with a vertical deviation of less than $0.25\mu\text{m}$.

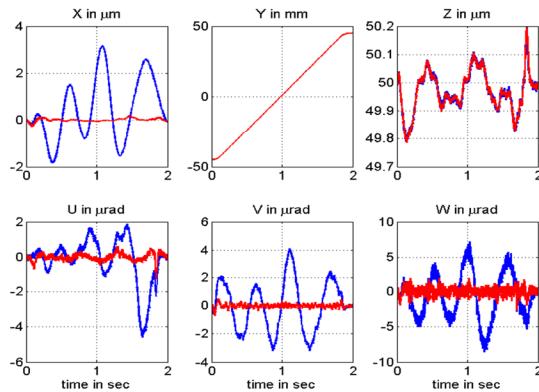


FIGURE 11. Horizontal movement with 50 mm/s without (blue curves) and with correction matrices (red curves)

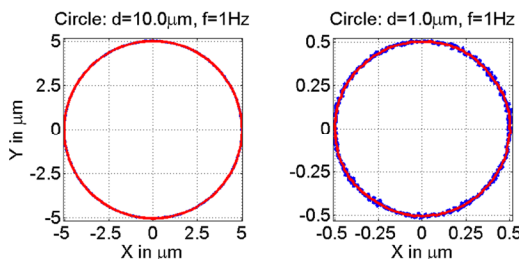


FIGURE 12. Circular motions in x - y -plane with diameters of $d=10\mu\text{m}$ and $d=1\mu\text{m}$ (red curves: $f_{\text{Pass}}=50\text{ Hz}$, $f_{\text{Stop}}=100\text{ Hz}$, Stopband: -40 dB , Filter phase free)

FIGURE 12 shows circular motions of 2 different diameters. The tracking accuracy is in nanometer level. Due to the z -travel of $100\mu\text{m}$ also Helix movements can be performed with some nm-resolution (FIGURE 13). There are no hysteresis effects observed.

The controller setup in the prototype limits the maximum velocity to 100 mm/s at the moment

but there are no physical limits caused by the structure.

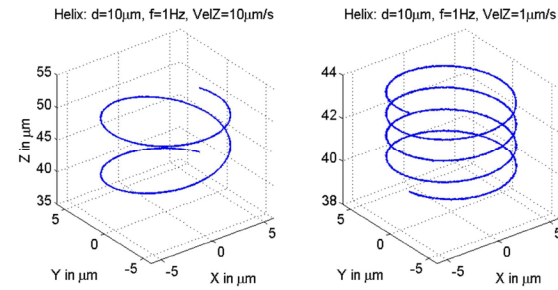


FIGURE 13. Helix movements with a diameter of $10\mu\text{m}$ and a z -velocity of $10\mu\text{m/s}$ and $1\mu\text{m/s}$

The high eigenfrequency of the platform assembly of more than 500 Hz allows high control bandwidth. The small noise level of all axes shown in FIGURE 10 opens the way for further improvements with new high resolution sensor systems in the servo loops.

CONCLUSION

A prototype of a 6 DOF planar magnetic levitation system with nanometer positioning capability as shown in FIGURE 3 has been constructed. It comprises an integrated compact 6D sensor and is controlled by PI's modular E-712 digital nanopositioning controller, using precalculated commutation functions for the three coil pairs realizing a smooth path control.

ACKNOWLEDGEMENT

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POSITIONING PERFORMANCE OF A PLANAR NANOPositionING SYSTEM WITH 100 MM CIRCULAR TRAVEL RANGE

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INTRODUCTION

High technology applications for example in the semiconductor or the optical industry require positioning systems providing repeatability and uncertainty in the range of nanometers together with x-, y- travel ranges of several hundreds of millimeters. Within the collaborative research center 622 „Nanopositioning and Nanomeasuring Machines“ (NPM/NMM) at the Ilmenau University of Technology scientists are working on providing the scientific basis for the necessary nanotechnological equipment [1]. We contribute in this research by investigating the applicability of integrated planar direct drives for the realization of NPM/NMM. Thereby we focus on the investigation and optimization of the drive system performance where ‘performance’ means the capability of the drive system to realize and hold a commanded position or path in closed loop operation.

The basic approach for the realization of NPM/NMM is a machine configuration with minimum abbe offset in the three translational axes. This is achieved by positioning a zerodur reflector relative to frame fixed interferometer heads [1]. Based on this principle, our approach is to use a planar direct drive system with aerostatic slider support for the lateral positioning of the zerodur reflector in a large travel range [2]. The integrated direct drive comprises three linear drive units, each consisting of a pair of frame-fixed flat coils and a permanent magnet array mounted on the sliders’ underside. The three drive units each generate a drive force acting on the slider and because of the 120° arrangement of the drive units a resulting force in any direction within the xy-plane can be generated. This is superposed by the generation of a torque around the z-axis for rotation control. The slider also carries plane mirrors so that its position can be measured precisely with a high-resolution single beam (x)

and a double beam (y, φ_z) plane mirror interferometer [3]. Thus, the movement of the slider can be controlled in x, y and φ_z . However, closed loop control is required in order to maintain the alignment of the reflectors and the interferometer heads as there is no mechanical fixture or guiding to prevent slider rotation.

PLANAR NANOPositionING SYSTEM

With this background a planar nanopositioning system was developed and realized where the slider is made of zerodur and has zerodur reflectors bonded to it. A photograph of the positioning system is shown in *FIGURE 1*. The key is that the slider of the integrated direct drive system and the reflector for the laser interferometers are incorporated in only one rigid moving body. This enables outstanding positioning capabilities. In the initial configuration the laser interferometers provided a measurement resolution of 0.1 nm and a servo error of $e_x=0.33$ nm and $e_y=0.45$ nm was realized in x and y [4]. Consequently the position quantization had to be considered to contribute to the remaining servo error. Together with the interferometer manufacturer SIOS the measurement resolution was reduced down to 0.02 nm for this experimental setup to improve performance and to examine the influence. The system now has the following characteristics:

- | | |
|---------------------------------------|-----------------------|
| • Travel range in x, y | Ø100 mm |
| • Resolution of measuring x, y | 0.02 nm |
| • Acceleration in x, y | 500 mm/s ² |
| • Positioning velocity | max. 30 mm/s |
| • Resolution of measuring φ_z | 0.8 nrad |
| • Moved mass | 9.6 kg |

For the realization of the closed loop control cascaded position- and rotation-controllers were designed and implemented on a rapid control prototyping hardware dSpace DS1006. On this

hardware the control algorithms are processed as a real-time application by a DSP-processor with 10 kHz sample frequency.

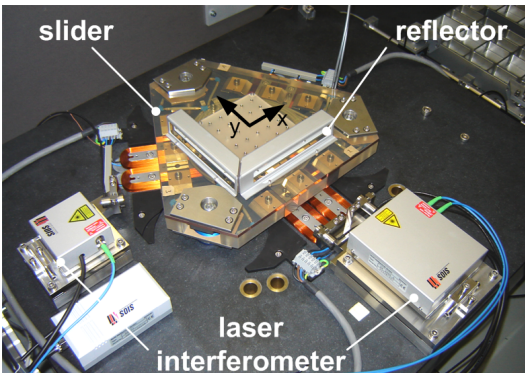


FIGURE 1. Planar nanopositioning system.

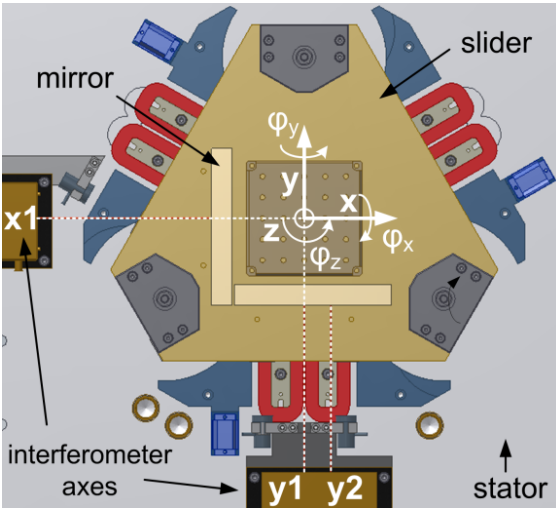


FIGURE 2. Measurement setup of the positioning system.

For a deterministic movement the slider is positioned with closed loop control where the x , y and ϕ_z coordinates are forced to the target value on the basis of the interferometric position and angle measurement, see FIGURE 2. So, for the x - and y -directions the straightness of motion is not defined by any mechanical guiding element but the machine coordinate system is defined by the 90° alignment of the two reflectors to each other while the straightness of motion along the axes is defined by the flatness of the mirror surface profiles. However, the z -position of the slider as well as the rotation around the horizontal axes ϕ_x and ϕ_y are not controlled in closed loop but determined by the flatness of the air bearing reference plane and the potential occurrence of parasitic vertical

forces at the drive units. The granite base is polished to a flatness of $1.0 \mu\text{m}$ in an area of $450 \times 450 \text{ mm}^2$. Given the long term stability of the $4 \mu\text{m}$ air gap, the travel of 100 mm leads to a worst case error motion in z of $1 \mu\text{m}$ while the worst case angular error motion is 1.8 arcsec in ϕ_x and 1.6 arcsec in ϕ_y respectively.

PERFORMANCE EVALUATION

Fixed environment setup

In a first step, the position of the slider was measured with a fixed environment, that is, with the air bearings turned off and the closed loop control not in operation. In this state there is a rather stiff connection between the reflector and the measuring head. The measured position signals are shown in FIGURE 3 and are referred to as fixed environment noise. It comprises actual position deviations originating from present disturbance vibrations as well as interferometer noise. The measured positions vary in a peak-to-peak range of $x_{pp}=1.4 \text{ nm}$ and $y_{pp}=1.2 \text{ nm}$. To characterize the present fixed environment noise the standard deviation was calculated for each of the 10 s time series: $\sigma_x = 0.17 \text{ nm}$, $\sigma_y = 0.16 \text{ nm}$. Compared to the former configuration with 0.1 nm measurement resolution the fixed environment noise reduced significantly, showing that the quantization made a significant contribution to the measured noise.

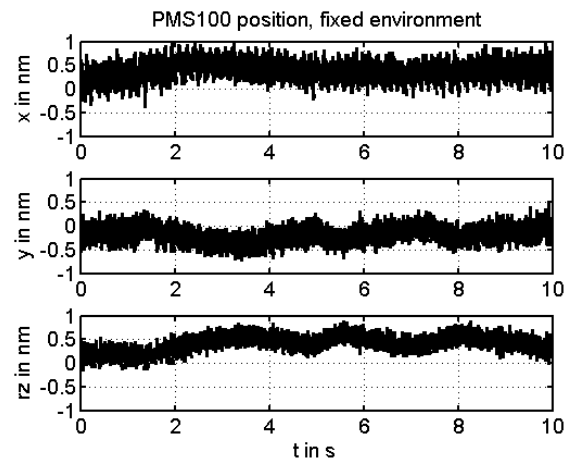


FIGURE 3. Measured position signals with fixed environment (primary values, 10 s trace)

Step and settle operation

With the position- and rotation-controller in operation a servo error of less than 0.5 nm is achieved in the center position. The measured position signals for x , y and ϕ_z for a target position $(x, y) = (0, 0)$ are shown in FIGURE 4. To quantify the drive systems' capability to

achieve the target position, the servo error was calculated as the root mean square (RMS) value of the position error. The servo error is as low as $e_{xRMS} = 0.22$ nm, $e_{yRMS} = 0.34$ nm, with peak-to-peak values: $e_{xPP} = 1.6$ nm, $e_{yPP} = 2.8$ nm. As 2D-positioning is the general case the resulting xy-servo error was calculated: $e_{xyRMS} = 0.29$ nm.

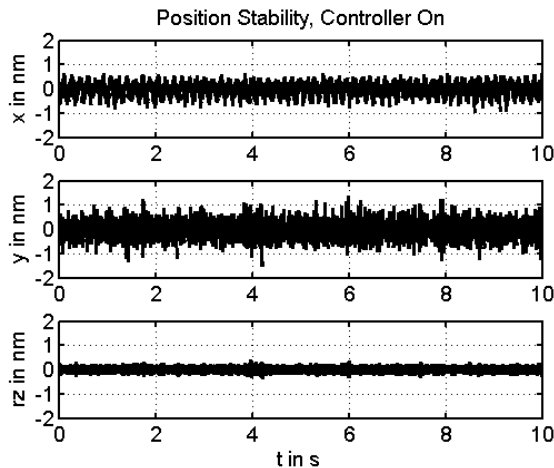


FIGURE 4. Position stability with lifted slider and controller turned on (primary values, 10 s trace)

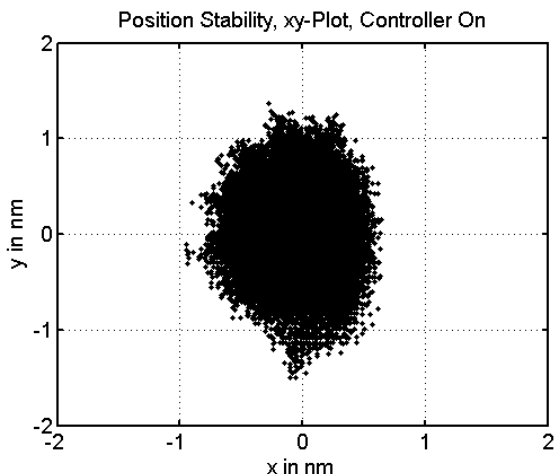


FIGURE 5. XY-plot of the measured position signals (primary values, 10 s trace)

These measurements were repeated at multiple positions within the $\varnothing 100$ mm travel range. Therefore the travel range was filled with a grid of measurement positions with 2 mm grid distance. At each position the slider stopped and the actual slider position was traced with 10 kHz for 1 s. From the time series again the RMS servo errors e_x , e_y and the resulting 2D servo error e_{xy} were calculated. FIGURE 6 shows the result of this measurement which took about 6 h.

The achieved xy-servo error varies between 0.3 and 1.3 nm. The 3D illustration of the spatial distribution indicates that the best performance is achieved in the center of the travel range while in the outer areas the servo error increases.

From these experiments we conclude that the drive system achieves a servo error of better than 1.3 nm within the whole travel range.

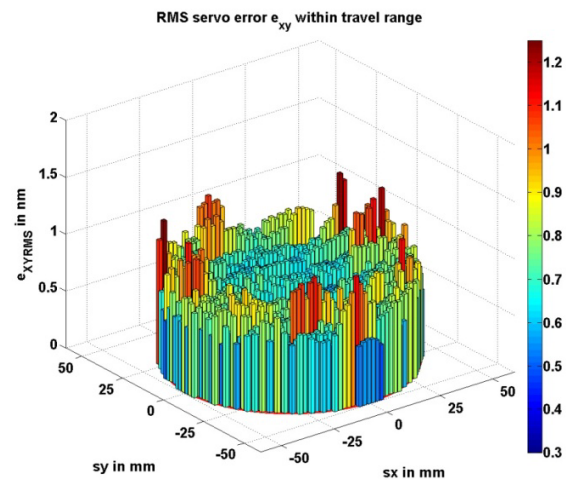


FIGURE 6. Spatial measurement of the 2D servo error within the travel range

The position stability within the positioning range gives an indication of the positioning repeatability. Addressing one position multiple times, the closed loop control will make the mean value of the actual position match the target position. However, for a certain point in time the actual position varies to the extent of the peak-to-peak servo error. In this way the repeatability of the system is related to the servo error. In the described experiment the maximum RMS servo error was 1.3 nm so that the position repeatability of the system is given as 3.9 nm.

Note in this context that the measurement uncertainty of the laser interferometric position measurements still adds an uncertainty to these values depending on the particular measurement conditions.

Drive System Resolution

An important performance measure is the resolution of the mechanical drive system. Although the laser interferometers provide a measurement resolution of 20 pm, this is not directly linked to the smallest step size the system can actually realize. In this context there are three aspects to mention:

(a) The present system noise (mechanical vibration, electric noise in the measurement system, etc.) is in the sub-nanometer range. In *FIGURE 3* a fixed environment measurement is presented where the standard deviations of the measured x , y time signals are below 0.2 nm.

(b) The digital controller (dSpace DS1006 hardware) internally processes the position values as 64 bit floating point numbers. At maximum travel, (e.g. at $x=50$ mm), the least significant bit represents a distance of 7×10^{-9} nm.

(c) The output variable of the control algorithm is a setpoint value for the coil current. Physically this is a voltage (± 10 V) put out with a 16 bit digital-to analog converter. It represents the driving voltage for the linear current amplifier. Thus, there is a lower limit for the incremental change of the coil current. However, this current increment is related to an incremental change of the drive force and does not necessarily determine the smallest position increment the system can perform.

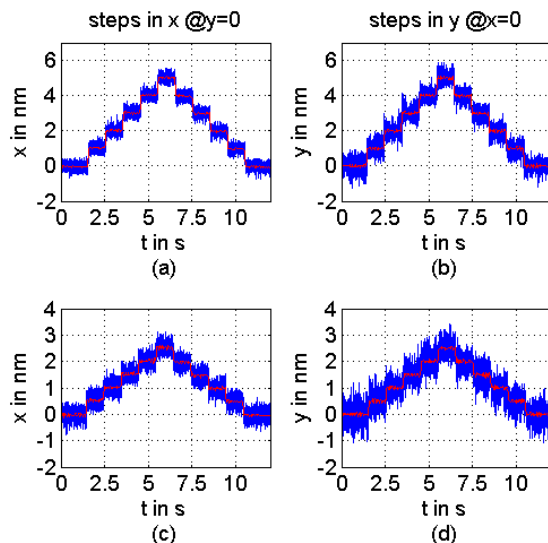


FIGURE 7. System step performance in x , y (a),(b): 1.0 nm steps; (c),(d): 0.5 nm steps

In fact it is rather the present noise that defines the smallest executable position change, which is considered to be clearly distinguishable from the noise band. *FIGURE 7* shows the measured values for stepwise positioning with step widths of 1.0 and 0.5 nm in x (a), (c) and y (b), (d). A filtered signal (500 sample running mean) is plotted together with the primary values. It can be seen that a step response is readily recognizable for the 0.5 nm steps.

Scanning Operation

Scanning operation in most applications means moving along a straight line with “constant velocity”. To test the scanning performance the system was commanded to move diagonally through the travel range with 10 mm/s. From the measured position values (9.5 s trace, 10 kHz sample frequency) a lateral position deviation from the straight line of $\sigma_{pl}=0.75$ nm (standard deviation) was derived. Along the straight line the movement is characterized by the actual velocity. It was calculated from the measured position values. A low pass filter (500 Hz cutoff, 2nd order) was applied to account for the misleading amplification of high frequency measurement system errors which results from the differentiation of the position values. The 500 Hz cutoff frequency thereby provides a sufficient distance to the crossover frequency of the position loop in the range of 100 – 120 Hz. As result a velocity standard deviation of 0.4 $\mu\text{m/s}$ was determined. In relation to the 10 mm/s command velocity this represents a velocity error <0.1 %.

CONCLUSION

A high precision planar positioning system was realized and an evaluation of the positioning performance has been carried out. The 2D RMS servo error is below 1.3 nm within the whole $\varnothing 100$ mm travel range and a positioning resolution of 0.5 nm has been proven in the center and in the boundary areas of the operating range. In scanning operation a velocity error of less than 0.1 ‰ was achieved.

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Ultrasonic Indenting of Nanofeatures

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INTRODUCTION

The use of anti-reflective (AR) coatings can be found in many consumer products available today. Due to growing global energy concerns, attention has turned to AR coatings to aid in the design of more efficient solar arrays and other energy harvesting technologies. While existing manufacturing techniques for these coatings tend to be costly and time-consuming, a novel approach to rapidly creating AR surfaces called nanocoining has been developed.

Nanocoining is a process in which a small die (20 x 20 μm) containing an array of 1600 nanofeatures is pressed into a diamond-turned surface, leaving a continuous field of the desired nanofeature impressions. Since the die covers only a small area, indentation must occur at high frequencies (50 KHz) to complete large areas in reasonable amounts of time. Spacing between indentations must be within 20 nm to maintain the uniform coverage of indents. FIGURE 1 is an SEM image of the nanofeatures on the die used for indenting.

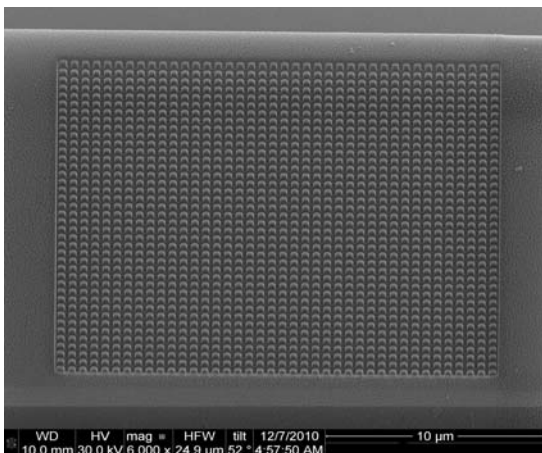


FIGURE 1. SEM image of nanofeatures on die surface.

The scope of this work is to create a uniform field of nanofeatures by developing an actuator capable of achieving the required speed and amplitude. This actuator will then be used to indent the radial surface of a precision-machined

drum at ultrasonic frequencies. Prior work has shown favorable results in 1 KHz indenting for producing nanofeatures on diamond-turned surfaces [1,2].

LINEAR ACTUATOR

Performing high speed indentations over large surface areas requires the use of an ultrasonic actuator to meet speed requirements. Large vertical amplitudes (on the order of 10 μm) are also desirable to minimize contact time between the die and indentation surface when the indent is being formed. Minimizing contact time is important in reducing smearing effects that occur due to a mismatch in surface velocities when the die contacts the rotating workpiece. As the vertical amplitude of the actuator increases, the contact time decreases for a given operating frequency.

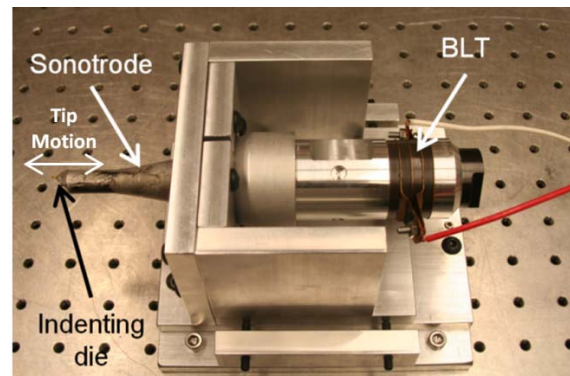


FIGURE 2. BLT/sonotrode setup in adjustable mount.

The first step was to use a 1D bolt-clamped Langevin type (BLT) transducer with an amplifying sonotrode horn to perform the operation (shown in FIGURE 2). Linear indentations were performed on the outer surface of a 1.5" diameter cylindrical workpiece using 2-axis DTM. As shown in FIGURE 3, the actuator was positioned normal to the workpiece surface and fed along the length of the cylinder.

Experimental results showed a non-continuous indent field on the workpiece surface. Analysis

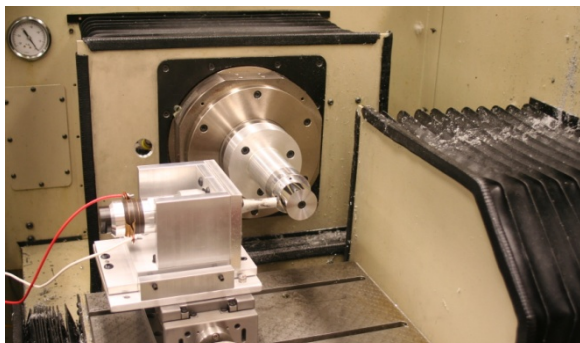


FIGURE 3. Ultrasonic indenting setup on diamond turning machine.

determined that due to low stiffness in the actuator mount, a 140 Hz resonant oscillation caused non-uniform indents. To mitigate low frequency affects, subsequent designs require high stiffness, low mass characteristics.

2-D ACTUATOR

Indent smearing can occur if the tangential speed of the die does not match the speed for the drum surface. To minimize these effects, a horizontal velocity component is added to the vertical indenting motion. Matching the horizontal motion component with the surface speed during each indent means that the die path must be elliptical or some other 2-D shape [3]. Achieving this elliptical die path, while maintaining sufficient amplitude at 50 KHz, requires a bidirectional resonant actuator. 2-D motion is created by combining two orthogonal mode shapes of the resonating actuator at the same frequency.

Concept

The actuator utilizes the vibration properties of a resonant beam to create the desired die motion at the tip. The beam geometry is sized such that the two vibration modes have the same natural frequencies, allowing for simultaneous excitation of both directions. Excitation occurs by using two piezoelectric elements that are fixed to opposing sides of the beam. The two piezo elements are excited at the single common mode frequency using amplified sine waves, causing contraction and expansion along the beam. By varying the phase between the two input signals, a combination of the two motions is measured at the actuator tip. A conceptual rendering is given in FIGURE 4.

The actuator is held in place at displacement nodes along the length of the beam. Node locations are chosen that are shared between

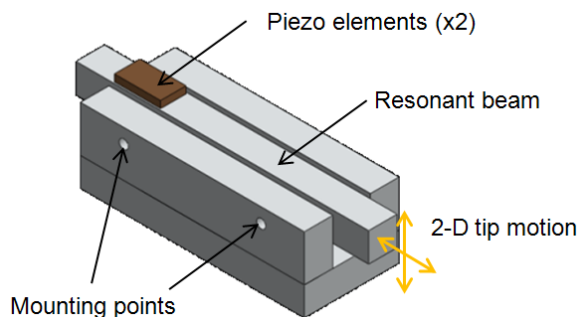


FIGURE 4. Resonant beam actuator concept.

the two operating mode shapes. Ensuring that the resonant structure is contacted only at mutual node locations is necessary to achieve the two dimensions of motion.

Design

The beam is designed to resonate in a longitudinal and bending mode, resulting in the desired 2-D tip motion. Matching the frequencies of these modes is done by designing the geometry of the actuator such that its longitudinal and bending modes resonate at the same frequency. Varying the geometry is also used to align nodes for mounting (FIGURE 5).

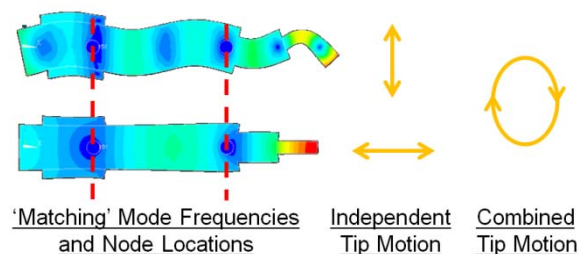


FIGURE 5. Longitudinal and bending modes resulting in elliptical motion at 49.7 KHz.

Analytical techniques provided initial approximations based on the desired scale and operation frequency of the actuator. Finite element analysis is then used to tune the geometry of the beam for matching frequencies. The variable cross-section design is a product of manipulating the geometry to align mode frequencies and node locations. It also serves to act as an amplifying-horn tip, while the axis-asymmetric design discourages out-of-plane motion. FIGURE 6 illustrates the final geometry and approximate dimensions.

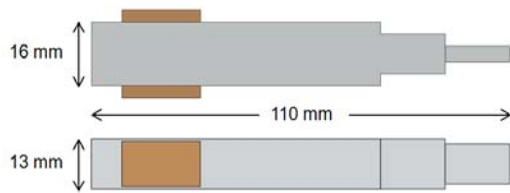


FIGURE 6. Resonant 2-D actuator geometry

Construction

The actuator is machined out of 6061-T6 aluminum to the model specifications. PZT elements (PKI 802) are sized and attached to the beam using conductive epoxy. Leads are soldered to the outer electrodes of both PZT elements, while the beam is treated as a common ground. The actuator mount sidewalls are machined to hold the mount screws in position to contact the node locations.

Testing

Tip displacement measurements were performed to determine the resonant frequency of both the longitudinal and bending modes. Transverse and longitudinal amplitude was simultaneously measured using a two channel photonic sensor (FIGURE 7).

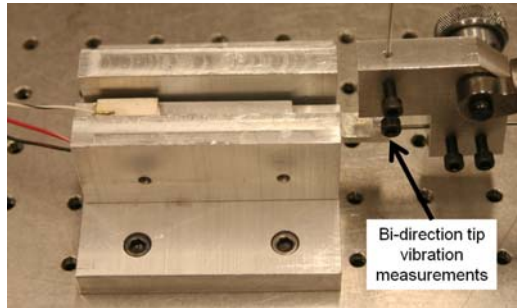


FIGURE 7. Experimental setup to measure longitudinal and transverse displacements.

The resonant frequency for each tip direction was measured independently by frequency sweeping to find the largest amplitude response. Longitudinal displacement measurements were achieved by exciting the two PZT elements in-phase with each other. Transverse measurements were taken by setting the two input signals 180° out-of-phase, resulting in pure bending actuation. The measured responses are given in FIGURE 8.

The tests to measure the resonant frequencies resulted in a bending mode of 48.1 KHz and a longitudinal mode of 48.4 KHz. While the

difference in the two frequencies is less than 1%, tuning must occur to ensure proper response from both modes at a single frequency.

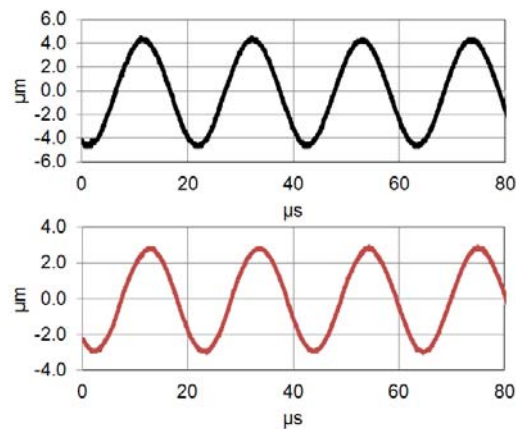


FIGURE 8. Longitudinal tip displacement from in-phase input signals (top). Transverse tip displacement from 180° out-of-phase input signals (bottom).

Tuning

Altering the resonant frequencies of the actuator requires the removal or addition of material to change the geometry of the beam. Comprehensive FEA simulations were performed to predict the effects of various dimensional changes to the design. The goal is to determine what alterations will impact the frequencies in the desired way.

The initial tests of the actuator revealed a bending mode that was 300 Hz lower than the longitudinal mode. Based on simulations, the structure was modified. A reduction in length of 0.9 mm resulted in matching bending and longitudinal frequencies at 48.57 KHz. A summary of these tests are given in TABLE 1.

TABLE 1. Measurements of resonant frequencies for change in actuator length.

	Initial	Test 1	Test 2	Test 3
Length (mm)	109.9	109.5	109.1	109.0
Bending (KHz)	48.10	48.30	48.45	48.57
Longitudinal (KHz)	48.40	48.50	48.54	48.57

With matching longitudinal and bending modes, the PZT elements can be excited 90° out-of-phase. This excites both modes simultaneously, resulting in an elliptical tip path (FIGURE 10).

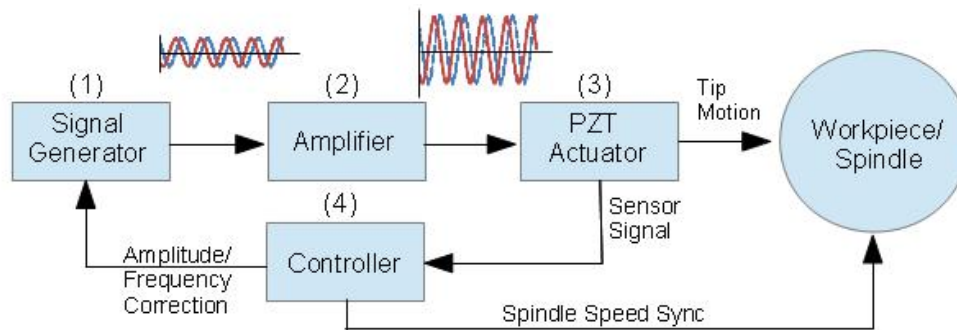


FIGURE 9. Schematic for closed-loop control for 2-D ultrasonic actuator.

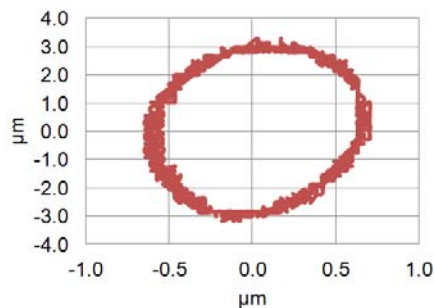


FIGURE 10. Elliptical tip path due to 90° out-of-phase input signals.

CONTROL CONSIDERATIONS

To produce large, uniform arrays of indents, the actuator must operate at resonance, with constant horizontal and vertical amplitudes. The control schematic is shown in FIGURE 10. Other work has shown viable closed-loop control strategies for multi-DOF ultrasonic actuators [4].

Frequency

The actuator must remain at resonance to avoid varying phase responses of the vibration modes. If phase shift occurs, the elliptical path could become distorted or reversed. Distortion of the die path causes poorly registered features due to excessive smear or changes in depth. The driving frequency is modulated to keep the actuator at its resonance.

Amplitude

The elliptical path dimensions are crucial in minimizing smear and maintaining a constant indentation depth. The vertical amplitude must remain constant to prevent the die indenting too deep or not contacting.

The horizontal dimension is critical in matching the workpiece's speed, as it dictates the lateral velocity of the die. A PZT sensor is used to monitor deflection of the actuator, and correct for variations in tip displacement. The voltage amplitudes are used to control the ellipse

dimensions, which must be constant within 100 nm.

Spindle Speed

As the resonant frequency changes, the speed of the workpiece must also change to maintain uniform indent spacing. The spindle must match the actuator operating frequency to keep registration position error of each indent to within one feature spacing, or about 100 nm.

CONCLUSIONS

Creating indentations at 30-50 KHz will be necessary for commercial acceptance of this fabrication technique. The key is the design of the actuator to produce the desired 2D die motion. An elliptical path is produced by the superposition of two orthogonal mode shapes at the same frequency. Frequency tuning and control capabilities are necessary in creating large, uniform areas of features.

ACKNOWLEDGEMENTS

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PLASMONIC NANO-LITHOGRAPHY MACHINE (PNLM)

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The goal of the current research project is to design, manufacture, assemble and calibrate a state of the art plasmonic imaging lithography machine. This project is a result of the cooperation between the “Center for Precision Metrology” (CPM) located at University of North Carolina at Charlotte and the Center for Scalable and Integrated NanoManufacturing (SINAM) located at University of California at Berkeley.

1. INTRODUCTION:

The lithography can be done using many techniques such as Photolithography (optical, UV, EUV), E-beam/ion-beam lithography, X-ray lithography, Interference lithograph and Nanoimprint. Imprint lithography and optical lithography are highly attractive due to the high throughput for mass fabrication. Also, photolithography has been a major fabrication method in the integrated circuit (IC) and semiconductor industries over the past several decades. Nanoimprint lithography has demonstrated great capabilities, however, the leveling of the imprint template and the substrate during the printing process is a challenge.

Deep UV photolithography can offer a resolution of sub-100 nm; but the spacing between patterns and the minimum feature size are determined by the diffraction limit of light [2]. In order to overcome the diffraction limit some technologies were developed such as near field interference lithography and phaseshifting mask lithography [2].

There are some critical concerns when generating subwavelength features from apertures smaller than the exposing wavelength such as low transmission through the apertures, poor contrast and limitation of the exposing distance [2].

2. PLASMONIC IMAGING LITHOGRAPHY (PIL)

Surface Plasmons are fluctuations in the electron density at the material boundaries and can strongly couple with photons. With this effect, some metal structure can “focus” light into a nano-spot in near-field. This structure is called Plasmonic Lens (PL) [3].

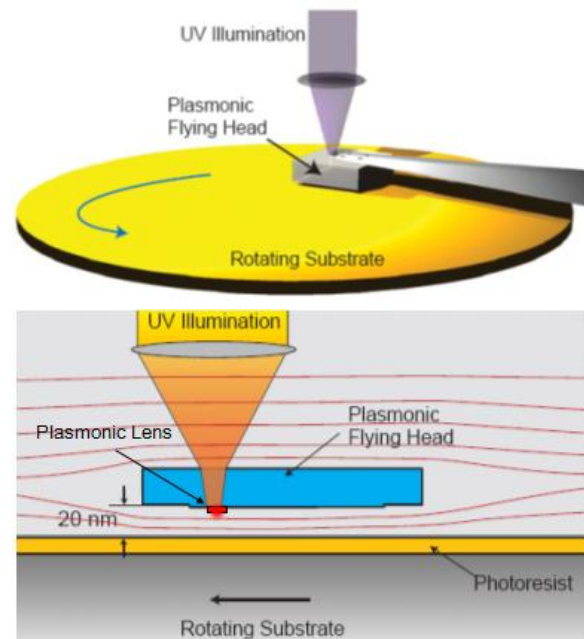


Figure1. Illustration of Plasmonic Lithography [3]

Plasmonic Imaging Lithography (PIL) has many advantages such as being high resolution, maskless, Lower cost, and high speed technique. Also, the PIL is highly flexible, user friendly, and has the potential to achieve throughputs that are two to five orders of magnitude higher than other maskless techniques.

3. PLASMONIC NANO LITHOGRAPHY MACHINE (PNLM) DESIGN

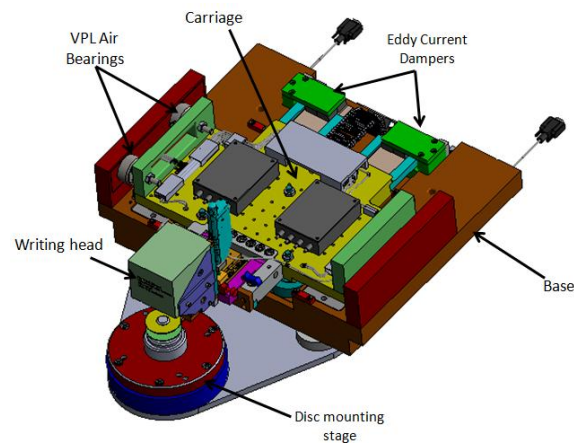


Figure 2. Plasmonic Imaging Lithography Machine (PNLM)

The machine composed of an aluminum base which stands on 3 leveling feet. The disc is mounted on 1000-3000 RPM rotary stage in order to provide a compensation for thermal expansion, an aluminum base plate has been designed where the disc mounting stage and the front two leveling feet will rest on.

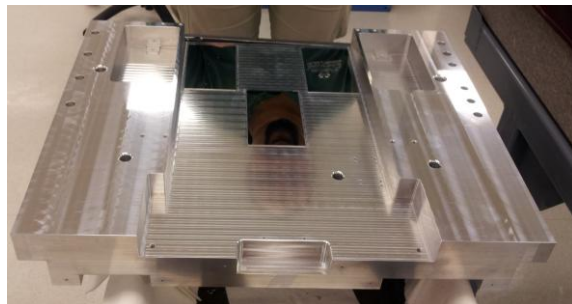


Figure 3. PNLM's base.

The carriage is made of AL 6061, and to provide a frictionless motion, three Vacuum Preload (VPL) airbearings are carrying the carriage weight. The linear motion needed to move the carriage is provided by a two three-phase Helbach linear motors. The Helbach array motor provides an accurate and smooth motion that can be controlled. The Helbach linear motor generates force in both longitudinal and vertical directions. Each motor composes of a 24-coils stator and an eight-magnets Helbach array.

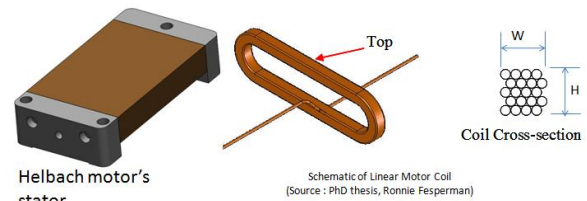


Figure 4. Illustration of the stator design of Helbach linear motor.

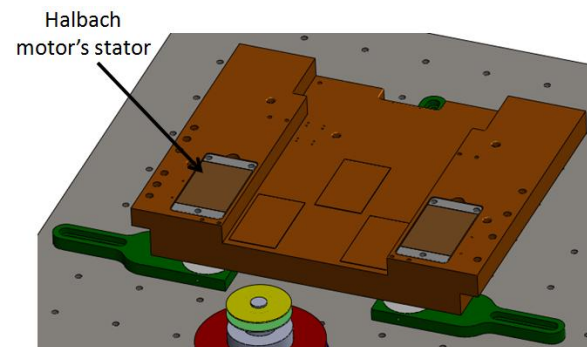


Figure 5. Illustration of the stators locations on the PNLM base.

In order to achieve the required accuracy, two 5-nanometer linear scales has been attached to the carriage. The read heads are mounted on the base, and the readings of these nano-precision scales can be used to determine the carriage position and to determine the actual linear speed. Also, the difference between the two readings can be used to determine any yaw that may happen during the linear motion and based on that a correction feedback signal will be generated.

Also, damping has been introduced to the system to provide a smooth linear motion. Eddy Current Dampers (ECD) have been designed and manufactured. The ECD composes of a copper plate that moves in between a magnet array composed of eight magnets. The yoke that holds the magnet array will be attached to the base and the copper plate will be attached to the carriage using an aluminum bracket. The carriage average linear speed is $5 \mu\text{m/s}$, and at this speed the ECDs will generate 0.01N damping force.

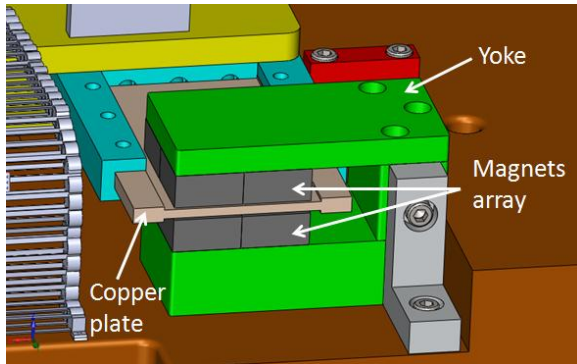


Figure 6. Illustration of the Eddy Current Damper (ECD) design.

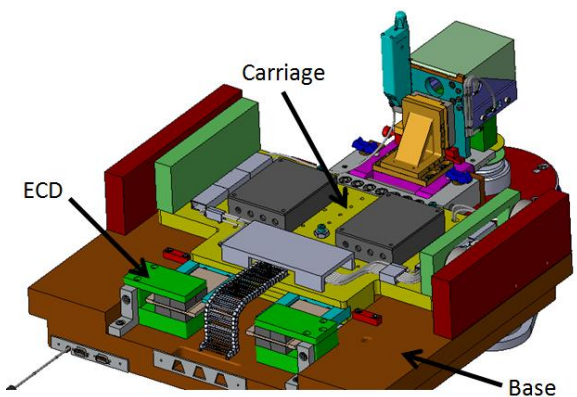


Figure 7. Illustration of the ECD locations on the PNLM.

The writing head is composed of a main slider, two brackets that hold the prefocusing stage, suspension bracket and the White Light Interferometer (WLI) bracket.

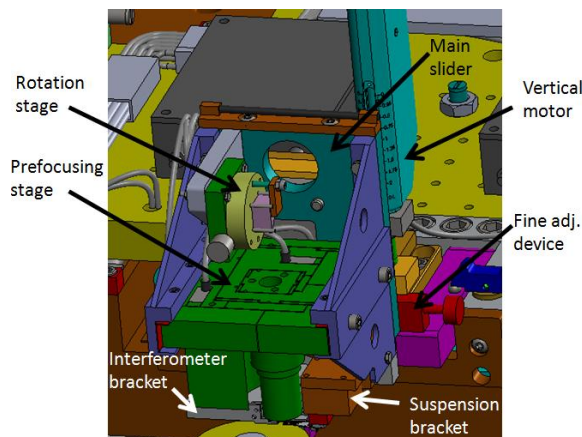


Figure 8. Illustration of the writing head components.

Also, the main slider is equipped with a 2" linear motor that provide a vertical motion of the writing

head during the plasmonic lens changing procedure. Also, the slider is resting on a fine adjustment device which is attached to a rotation arm that will provide a 90° rotation needed during the PL changing procedure.

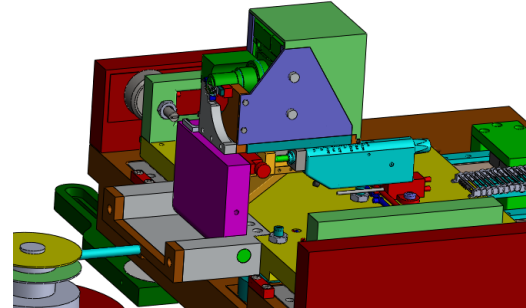


Figure 9. Illustration of the writing head rotation during the PL changing procedure.

The PL can move in the radial (downtrack) and tangential (off track) directions using piezoelectric shear actuator of 3μm range in both directions. Also, the machine is equipped with a White Light Interferometer (WLI) attached to the writing head. Christopher Hughes designed the WLI bracket and he is working on the interferometer design. Christopher uses two needles as fiber optics holders and directed the laser beams to mirrors attached to the slider sides. After measuring the vibration, a feedback signal will be generated and applied on a two dimensional shear actuator to make any required correction.

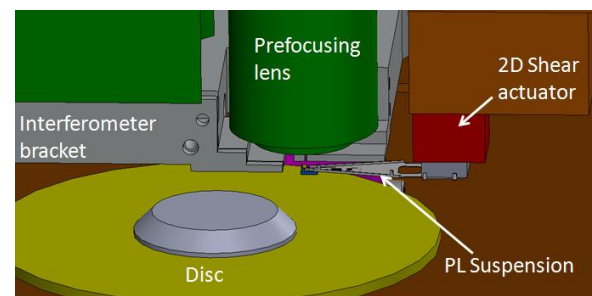


Figure 10. Illustration of the PL and the shear actuator locations.

And to provide the required prefocusing of the UV beam, a prefocusing stage has been designed by Dr. Wen Wang. The prefocusing stage provides three dimensional motion of the prefocusing lens for a 50x50x100 μm range. Also, the writing head is equipped with a rotation stage to direct the UV beam to the prefocusing lens.

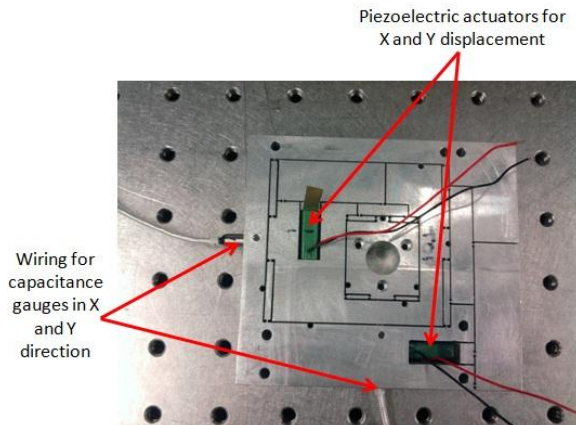


Figure 11. The prefocusing flexure designed by Dr. Wen Wang.

4. THE PNLM SPECIFICATIONS:

The PNLM is designed to produce 45 nm line width at 45 nm half pitch. The overlay (3σ) is expected to be 11 nm (1/4 of pixel size). The following figure illustrates the machine coordinate system.

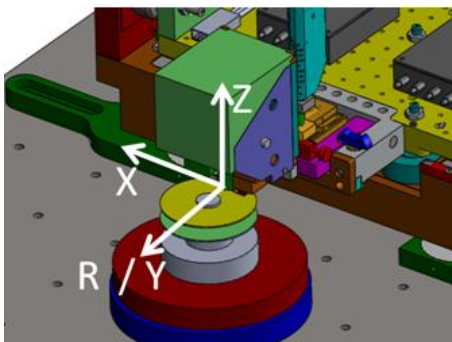


Figure 12. Coordinate system on the PNLM

The PL maximum travel range is (2.26") 57.40 mm in the radial direction, this will allow for future developments of the machine. Currently, the requirement is set to be only (1.5") 38.1 mm travel length.

A tabulated list of the PNLM performance specifications is shown below.

Table 1. PNLM Performance Specifications.

Maximum Radial (Downtrack) Travel	57.40 mm (2.26")
Radial Travel	38.1mm (1.5")
Radial Position Error	5 nm
Radial Resolution	< 5 nm

Radial Minimum Speed	100 nm/s
Radial Average Speed	5 $\mu\text{m/sec}$
Pixel Size	45 nm
Accuracy	5 nm
Prefocusing Stage range	X and Y :50 μm Z :100 μm
Disk spinning speed limits	1000-1500 RPM
Average Writing time (full disk)	5 hrs
PL Lifetime (pre-alignment)	2 min
PL Lifetime (post-alignment)	20 min
Carriage Angular Stiffness	75 N/ $\mu\text{-rad}$
Carriage Linear (lateral) stiffness	26 N/ μm
Carriage Linear (vertical) stiffness	39 N/ μm
θ_x Leveling Error	1 $\mu\text{-rad}$
Z-axis Fine Motion Travel	6mm
Z-axis Coarse Travel (Vertical motor)	50.8 mm (2")
Alignment Time	1 min

5. FUTURE WORK:

The machine parts have been designed and analyzed statically and dynamically using a design software package. The machine is currently in the assembly stage. After assembly, the machine will be tested and calibrated and any necessary adjustments will be made, then the machine will be shipped to SINAM's lab at the University of California at Berkeley.

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PROGRESS IN THE DEVELOPMENT OF PRECISION POSITIONING STAGES FOR X-RAY SCANNING NANOPROBES

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INTRODUCTION

The Advanced Photon Source (APS) at Argonne National Laboratory (ANL) is a national user facility for synchrotron radiation research. The high-brilliance x-ray beams of this third-generation synchrotron radiation source provide powerful tools for forefront basic and applied scientific research, including state of the art x-ray microscopes, x-ray microprobes and x-ray nanoprobes. Precision positioning techniques present a significant opportunity in support of these synchrotron radiation instruments.

The APS has developed the hard x-ray nanoprobes beamline at APS Sector 26, which has been operated, in partnership with the Center for Nanoscale Materials (CNM), since 2008. This system covers an energy range of 3-30 keV, and utilizes diffraction, fluorescence, and full-field transmission imaging to enable the study of nanoscale materials and devices. The nanoprobes have to date achieved a spatial resolution of 40-nm in scanning mode, and of 30-nm in full field mode [1-3]. The high spatial resolution is enabled by a combination of a high-resolution positioning system based on flexural stages, and high accuracy provided by laser interferometer-based encoders.

Inside the instrument chamber, as shown in Figure 1, the positioning stages for the optics and specimen are grouped into several subcomponents, the condensor module (CM) (1), the focusing optics module (FOM) (2), the specimen module (SM) (3), and the imaging optics module (IOM) (4). These modules are engineered to be moved in or out of the beam to allow configuration of the instrument for either scanning mode or full-field mode. A specially designed, custom-built laser Doppler displacement meter system (LDDM) provides two-dimensional (2-D) differential displacement measurements with subnanometer resolution between the zone-plate x-ray optics and the sample holder. A total of six LDDMs is applied

for the positioning stage system to correct possible linear motion trajectory errors [3].

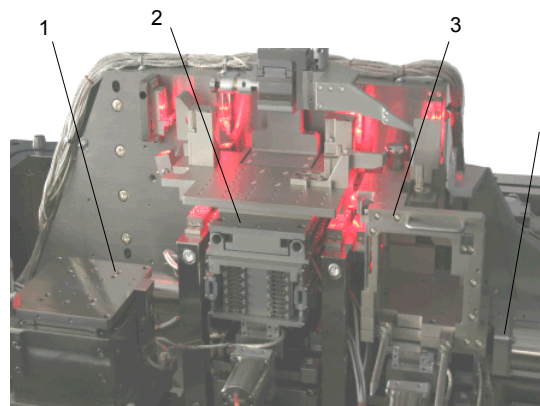


FIG. 1. A photograph of the positioning stages for optics and specimen manipulation inside the CNM/APS hard x-ray scanning nanoprobes instrument chamber. 1: CM; 2: FOM; 3: SM; 4: IOM.

Based on the experiences gained from the CNM/APS hard x-ray nanoprobes design, construction, and operation, further efforts have been made to develop new precision positioning stages for APS users and for the new *in situ* hard x-ray nanoprobes and sub-micron 3-D diffraction beamlines for the APS Upgrade project [4]. The development efforts are focused in the following areas:

- Multi-dimensional tip-tilting flexural stages for hard x-ray nanometer-scale 2-D focusing optics with Kirkpatrick-Baez (K-B) or nested K-B mirrors [5];
- Vertical and horizontal linear flexural stages with 2-kg load capacity, 3-mm (V) x 3-mm (H) travel range, and sub-nanometer positioning resolution for x-ray nanoprobes sample manipulations [6,7];
- Precision multi-dimensional alignment apparatus for multilayer Laue lenses

(MLLs) with nanometer-scale 2-D focusing [8];

- Customized high-stiffness vertical and horizontal linear stages with 30-kg load capacity, 12-mm (V) x 20-mm (H) travel range, and 20-nm positioning resolution for x-ray nanoprobe sample manipulations;
- Advanced laser interferometric sensor systems for linear and angular displacement measurements [9].

Progress in development of these precision positioning stage systems is summarized briefly in this paper.

TIP-TILTING FLEXURAL STAGES

High-stiffness rotary weak-link mechanisms based on stacked thin-metal sheets were first developed for the APS high-energy-resolution beamline 3-ID. Using a laminar structure configured and manufactured by chemical etching and lithography techniques, we were able to design and build a planar-shape, high-stiffness, high-precision, weak-link module. The precision and stability of this mechanism allowed us to align or adjust an assembly of crystals to achieve the same performance as that of a single channel-cut crystal. More than 50 sets of such rotary weak-link mechanisms have been made for APS users, with applications including high-energy-resolution monochromators for inelastic x-ray scattering, x-ray analyzers for ultra-small-angle scattering, and diffracted-beam crystal analyzers for powder-diffraction experiments. Their typical angular positioning resolution is 10-30 nrad with a travel range of up to 1.2 degrees [10].

To expand the applications of these rotary weak-link mechanisms to vertical axis configurations, linear weak-link modules have been designed and incorporated into a new flexural stage system for vertical axis and tip-tilting stages applications. A prototype of the two-dimensional tip-tilting stage system has been designed and constructed as shown in Figure 2. It is designed for a novel nested prefigured elliptical mirror system (Montel x-ray mirror system) to push the performance of polychromatic microdiffraction far beyond current levels at the APS Sector 34 [5]. The goal of this effort is to provide an ultra-stable mirror-based x-ray nanofocusing system

for a hard x-ray nanoprobe and an x-ray micro-diffraction instrument for high-resolution images of 3-D crystal structures.

As shown in Figure 3, the two-dimensional tip-tilting flexural stage system consists of a vertical-axis rotary weak-link top stage and a horizontal-axis tip-tilting weak-link base stage. To enhance the load capacity and stiffness in a vertical axis configuration, six linear weak-link modules are mounted on the sides of the base plate and top plate, orthogonal to the rotary weak-link plan and act as a vertical stiffener for the vertical axis [5]. With this new design, 2-kg or higher vertical load capacity is achievable. Table 1 shows the design specifications for the APS Z8-53 two-dimensional tip-tilting flexural stage system.

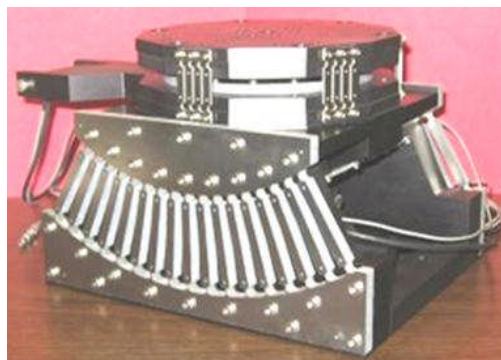


FIG. 2. A photograph of the prototype of the two-dimensional tip-tilting stage system for APS 34-ID.

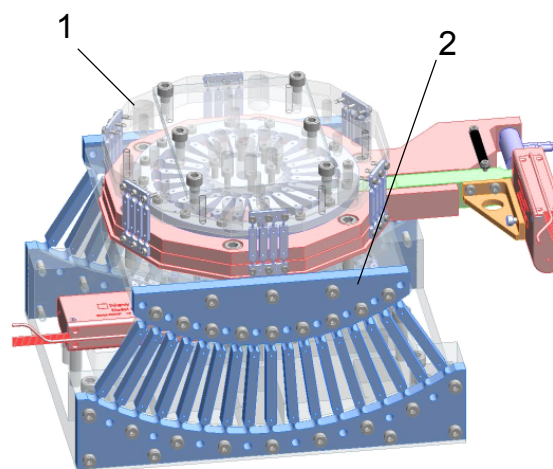


FIG. 3. A 3-D model of the two-dimensional tip-tilting flexural stage system: (1) Vertical-axis rotary weak-link top stage; (2) Horizontal-axis tip-tilting weak-link base stage.

Table 1. Design specifications for the APS Z8-53 two-dimensional tip-tilting flexural stage system.

Vertical-axis rotary top stage	
Overall dimensions (mm)	236 (L) x 158 (W) x 50 (H)
Normal load capacity (kg)	2
Driver type	PZT and Picomotor™
Encoder type	Strain sensor/optical sensor
Angular travel range (mrad)	26
Min. incremental (nrad)	10
Horizontal-axis tip-tilt base stage	
Overall dimensions (mm)	212 (L) x 184 (W) x 108 (H)
Normal load capacity (kg)	5
Driver type	PZT and Picomotor™
Encoder type	Strain sensor/optical sensor
Angular travel range (mrad)	26
Min. incremental (nrad)	10

VERTICAL AND HORIZONTAL LINEAR FLEXURAL STAGES

Ball-bearing-based or roller-bearing-based linear positioning stages are capable of providing a larger travel range. However, their positioning resolution is limited by the friction of the bearing-based guiding system. Moreover, their linear motion straightness of trajectory usually is not repeatable in nanometer scale due to the roundness errors of the bearing's rolling element and the uncertainty of the rolling ball's or roller's sliding effect. In the travel range of less than a few hundred microns, commercial PZT-driven flexural linear stages are available, but, it was hard to find any suitable for a travel range larger than few millimeters with high stiffness and load capacity.

In 2008, the linear vertical/horizontal flexural stages APS T8-31/32, with 3-mm travel range using fishbone-shaped multiple parallelogram weak-link structures, were developed at the APS at ANL for nanopositioning of a specimen holder for an MLL-based hard x-ray nanofocusing test bed at APS Sector 26 [6]. The T8-31 and T8-32 flexural stages were driven by PI Nexline™ N-214 and N-110 PZT actuators.

To extend the stage driving force and load capacity, T8-33 and T8-34 stages were designed with PI Nexline™ N-216 and N-111 PZT actuators. Two sets of T8-33/34 stages were constructed in 2010, one for the APS Sector 2 hard x-ray microscope station with high vacuum compatibility, and one for the NSLS-II MLL-based hard x-ray nanoprobe test-bed installed at Diamond Light Source (DLS), UK [7]. To improve closed-loop positioning repeatability, external capacitive or optical sensors were

added on the APS T8-33/34 stages as shown in Figure 4. Figure 5 shows a 3-D model of the linear vertical/horizontal flexural stages T8-33/34. Table 2 summarizes the design specifications for the linear vertical/horizontal flexural stages T8-33/34.



FIG. 4. A photograph of the APS T8-33/34 sample stages group installed at DLS, UK.

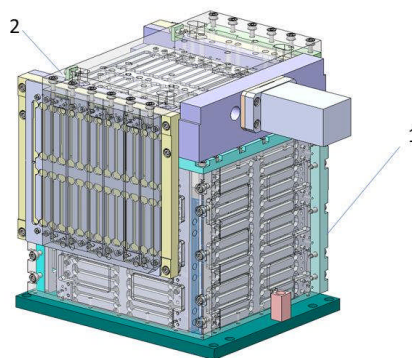


FIG. 5. A 3-D model of the linear vertical/horizontal flexural stages APS T8-33/34: (1) Vertical base stage T8-33; (2) Horizontal top stage T8-34.

Table 2. Design specifications for the APS T8-33/34 linear vertical/horizontal flexural stages system.

Vertical base stage T8-33	
Overall dimensions (mm)	174 (L) x 135 (W) x 139 (H)
Travel range (mm)	3
Max. load capacity (kg)	4
Min. incremental motion (nm)	1
Driving mechanism	PI Nexline N-216™
Encoder type	External capacitive or optical sensor
Stage weight (kg)	5
Horizontal top stage T8-34	
Overall dimensions (mm)	218 (L) x 180 (W) x 121 (H)
Travel range (mm)	3
Max. load capacity (kg)	2
Min. incremental motion (nm)	1
Driving mechanism	PI Nexline N-111™
Encoder type	External capacitive or optical sensor
Stage weight (kg)	2

PRECISION MULTI-DIMENSIONAL ALIGNMENT APPARATUS

The first prototype of precision multi-dimensional alignment apparatus for 2-D MLL x-ray focusing was designed and constructed for an MLL test-

bed at APS beamline 26-ID in 2008 [8]. With this test-bed, MLL demonstrated the focusing capability of hard x-rays to a 2-D focus of 25 nm horizontal x 27 nm vertical FWHM at a photon energy of 12 keV, and of 25 nm horizontal x 40 nm vertical FWHM at a photon energy of 19.52 keV [11]. In 2010, a second MLL test-bed was designed at the ANL for the National Synchrotron Light Source-II (NSLS-II) nanopositioning development project at Brookhaven National Laboratory (BNL). This work was carried out as collaboration between Argonne and Brookhaven scientists and engineers, in an effort to achieve NSLS-II's R&D goals for achieving 10-nm focusing of hard x-rays [7].

To perform ideal 2-D focusing, the MLL test-bed needs to provide the capability to align precisely two MLLs with respect to each other in a total of eight degrees of freedom. In the apparatus, the downstream MLL lens is mounted on the upper group of stages that provides two translational and one rotational degree of motion. The upstream MLL lens is mounted on the lower group of stages that provides precision positioning in five degrees of freedom, including three translational and two angular adjustments.

Figure 6 is a photograph of the alignment apparatus, and Figure 7 shows a 3-D model of the multidimensional alignment apparatus for x-ray MLLs with nanometer-scale 2-D focusing.

The second MLL test-bed has been installed, and commissioning began at a long imaging beamline experimental station (I13L) at the Diamond Light Source (DLS), UK in April 2012. It will be used as an initial platform for further nanofocusing developments in collaboration with the DLS staff [12].



FIG. 6. A photograph of the multidimensional alignment apparatus for the x-ray MLLs installed at the long imaging beamline experimental station (I13L) at the Diamond Light Source (DLS), UK.

SUMMARY

Progress has been made to develop new precision positioning stages for APS users and for the new in situ hard x-ray nanoprobe and sub-micron 3-D diffraction beamlines for the APS Upgrade project. Nanopositioning techniques present a significant opportunity for supporting the state-of-the-art synchrotron radiation instrumentation research at the APS Upgrade project.

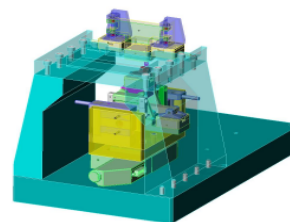


FIG. 7. A 3-D model of the multidimensional alignment apparatus for x-ray MLLs with nanometer-scale 2-D focusing.

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Keywords: x-ray nanoprobe, nanopositioning, precision stage.

Process Damping Analytical Stability Analysis and Validation

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INTRODUCTION

The analytical stability lobe diagram offers a predictive capability for selecting stable chip width-spindle speed combinations in machining operations [1-4]. However, the increase in allowable chip width provided at spindle speeds near integer fractions of the system's dominant natural frequency is diminished substantially at low spindle speeds where the stability lobes are closely spaced. Fortunately, the process damping effect can serve to increase the chatter-free chip widths at these low speeds. This increased stability at low spindle speeds is particularly important for hard-to-machine materials that cannot take advantage of the higher speed stability zones due to prohibitive tool wear at high cutting speeds.

This paper presents an analytical solution for milling stability that includes process damping effects. A velocity-dependent process damping force model, which relies on a single coefficient, is applied. The process damping coefficient is identified experimentally for a selected tool-workpiece pair and the effects of tool wear and relief angle are evaluated.

PROCESS DAMPING DESCRIPTION

Interference between the cutting tool's flank face and the undulations left behind on the cut surface serves as an energy dissipation mechanism and increases stability at low cutting speeds. This energy dissipation at the tool/workpiece interface is recognized as the main source of process damping. The process damping force, F_d , in the surface normal (n) direction can be expressed as a function of cutter velocity, $\dot{n}$, chip width, b , cutting speed, V , and a process damping coefficient, C [5]. See Eq. 1.

$$F_d = -C \frac{b}{V} \dot{n} \quad (1)$$

This process damping force, which depends on both the spindle speed-dependent limiting chip width and the cutting speed, is not included in

traditional regenerative chatter analytical solutions. Therefore, an iterative solution that includes process damping effects was developed. The empirical process damping coefficient is the only value required to describe the process damping model. This is analogous to the specific cutting force, K_s , approach used to model cutting force.

STABILITY ALGORITHM

The process damping force model defined in Eq.1 is applied for an up milling example. The geometry is shown in Fig. 1, where n is the surface normal direction. The projection of the process damping force from the n direction onto the x direction is:

$$F_x = F_d \cos(90 - \phi_{ave}) = -\left(C \frac{b}{V} \cos(90 - \phi_{ave})\right) \dot{n}. \quad (2)$$

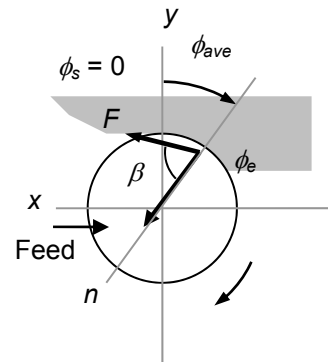


FIGURE 1. Model for up milling (a 50% radial immersion cut is shown). The line marked n defines the average surface normal direction.

Substituting $\dot{n} = \cos(90 - \phi_{ave}) \dot{x}$ in Eq. 2 gives:

$$F_x = -\left(C \frac{b}{V} \cos^2(90 - \phi_{ave})\right) \dot{x}. \quad (3)$$

The new damping in the converging stability calculation for the x direction frequency response function, G_x , is therefore:

$$c_{new,x} = c_x + C \frac{b}{V} \cos^2(90 - \phi_{ave}). \quad (4)$$

Note that c_x is the original damping from the system structural dynamics. The new y direction damping is:

$$c_{new,y} = c_y + C \frac{b}{V} \cos^2(180 - \phi_{ave}). \quad (5)$$

Replacing the original structural damping with c_{new} in the system's frequency response function enables process damping to be incorporated in the traditional regenerative chatter stability analysis.

Thusty's time invariant stability model for milling [3] is used to develop analytical stability lobes for the up milling example. Because the new damping value is a function of both the spindle speed-dependent limiting chip width and the cutting speed, the b and Ω vectors must be known in order to implement the new damping value. This leads to the converging nature of the stability analysis that incorporates process damping. The following steps are completed for each lobe in the stability lobe diagram:

1. the analytical stability boundary is calculated with no process damping to identify initial b and Ω vectors
2. these vectors are used to determine the corresponding c_{new} vector
3. the stability analysis is repeated with the new damping value to determine updated b and Ω vectors
4. the process is repeated until the stability boundary converges.

To demonstrate the approach, consider the model in Fig. 1 with $k = 9 \times 10^6 \text{ N/m}$ (stiffness), $f_n = 900 \text{ Hz}$ (natural frequency), $\zeta = 0.03$ (damping ratio), $K_s = 2000 \times 10^6 \text{ N/m}^2$ (specific cutting force), $\beta = 70 \text{ deg}$ (force angle), and $d = 19 \text{ mm}$ (tool diameter). The stability boundary with no process damping ($C = 0$) is shown in Fig. 2 for 60 lobes. It is observed that the limiting chip width approaches the asymptotic stability limit of 0.50 mm for spindle speeds below 1000 rpm.

Results for the converging procedure with process damping are provided in Fig. 3. Converging behavior is observed for the 10 iterations as the lobes move up and slightly to

the right. Although a convergence criterion, such as a threshold percent difference between subsequent minimum values, could be implemented, a practical selection of 20 iterations was applied for the diagrams in this study to ensure convergence. Figure 4 displays the new stability diagram for 60 lobes with $C = 3.0 \times 10^5 \text{ N/m}$.

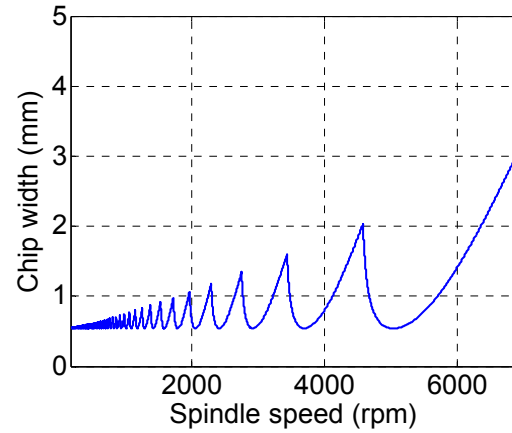


FIGURE 2. Stability diagram for symmetric up milling model from Fig. 1 with $k = 9 \times 10^6 \text{ N/m}$, $f_n = 900 \text{ Hz}$, $\zeta = 0.03$, $K_s = 2000 \times 10^6 \text{ N/m}^2$, $\beta = 70 \text{ deg}$, $d = 19 \text{ mm}$ and $C = 0$.

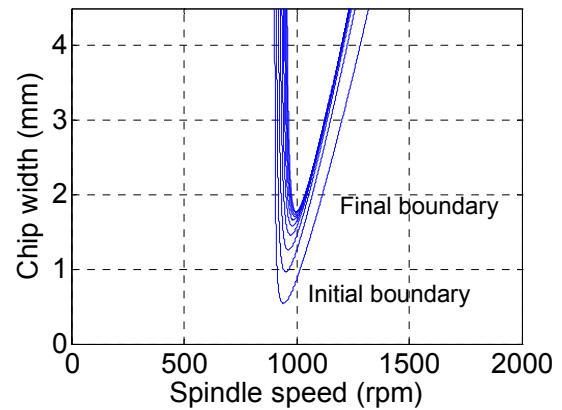


FIGURE 3. Convergence demonstration (10 iterations) for symmetric up milling model from Fig. 1 with $k = 9 \times 10^6 \text{ N/m}$, $f_n = 900 \text{ Hz}$, $\zeta = 0.03$, $K_s = 2000 \times 10^6 \text{ N/m}^2$, $\beta = 70 \text{ deg}$, $d = 19 \text{ mm}$ and $C = 3.0 \times 10^5 \text{ N/m}$.

EXPERIMENTAL VERIFICATION

Setup Description

In order to provide convenient control of the system dynamics, a single degree-of-freedom, parallelogram leaf-type flexure was constructed

to provide a flexible foundation for individual AISI 1018 steel workpieces; see Fig 5. Because the flexure compliance was much higher than the tool-holder-spindle-machine, the stability analysis was completed using only the flexure's dynamic properties. A radial immersion of 50% and a feed per tooth of 0.05 mm/tooth was used for all conventional (up) milling tests.

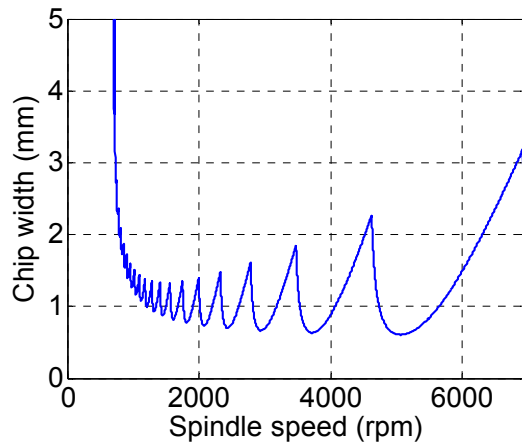


FIGURE 4. Stability diagram for symmetric up milling model from Fig. 1 with $k = 9 \times 10^6 \text{ N/m}$, $f_n = 900 \text{ Hz}$, $\zeta = 0.03$, $K_s = 2000 \times 10^6 \text{ N/m}^2$, $\beta = 70 \text{ deg}$, $d = 19 \text{ mm}$ and $C = 3.0 \times 10^5 \text{ N/m}$.

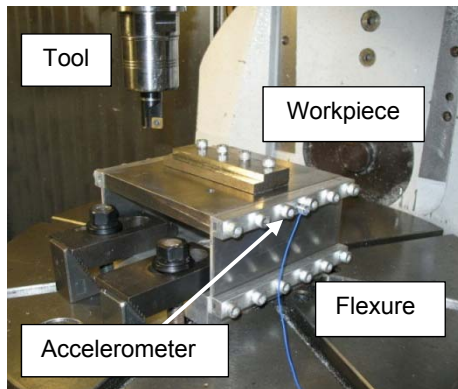


FIGURE 5. Setup for milling stability tests. An accelerometer was used to measure the vibration signal during cutting.

An accelerometer (PCB Piezotronics model 352B10) was used to measure the vibration during cutting. The frequency content of the accelerometer signal was used in combination with the machined surface finish to establish stable/unstable performance, i.e., cuts that exhibited significant frequency content at the

flexure's compliant direction natural frequency, rather than the tooth passing frequency and its harmonics, were considered to be unstable.

The effect of the cutting insert relief angle was examined using 15 and 11 deg relief angle inserts. Wear effects were evaluated by milling with new and moderately worn inserts. The flexure dynamics were also adjusted to determine the sensitivity of the process damping coefficient to changes in the system dynamics.

Process Damping Coefficient Identification

A grid of test points at low spindle speeds was selected to investigate the process damping behavior. Based on the stable/unstable cutting test results, a single variable residual sum of squares (RSS) estimation was applied to identify the process damping coefficient that best represented the experimental limiting axial depth of cut, b_{lim} . The results of the coefficient identification method are depicted in Figure 6.

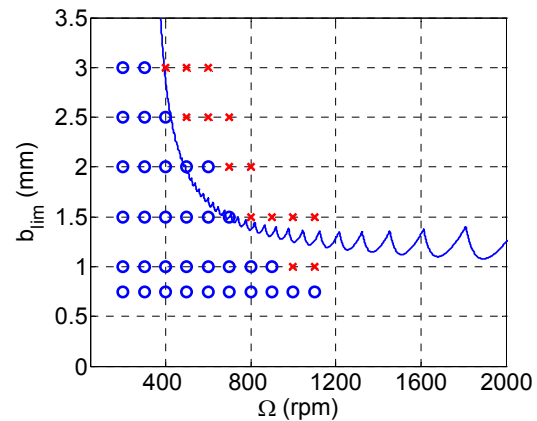


FIGURE 6. Up milling stability boundary for 50% radial immersion, 15 deg relief angle, low wear milling tests using the 228 Hz flexure setup ($C = 2.5 \times 10^5 \text{ N/m}$).

RESULTS

Cutting Tests

For a 50% radial immersion up milling cut, the stability boundary was discovered to increase as the spindle speed decreased and a value of for the process damping coefficient was obtained. Replacing the original 15 deg relief angle insert with an 11 deg relief angle insert and repeating the cutting tests revealed that the process damping coefficient increased by approximately 30% under low wear conditions. This indicates that the decreased relief face angle increases

process damping. The results for the low wear tests are summarized in Table 1.

TABLE 1. Comparison of process damping coefficients for low wear tests.

Relief angle (deg)	C (N/m) for the 228 Hz setup	C (N/m) for the 156 Hz setup
15	2.5×10^5	2.6×10^5
11	3.3×10^5	3.3×10^5

Tests were also completed using moderately worn inserts. Under worn conditions, the process damping coefficient increased by approximately 15% compared to new conditions. The process damping values obtained using moderately worn inserts are presented in Table 2. Finally, the dynamic properties of the system were adjusted by increasing the mass fixed to the flexure. No appreciable change in the process damping coefficient was observed.

TABLE 2. Comparison of process damping coefficients for moderate wear tests.

Relief angle (deg)	C (N/m) for the 228 Hz setup	C (N/m) for the 156 Hz setup
15	3.0×10^5	3.4×10^5
11	4.0×10^5	3.8×10^5

Repeatability

Repeat testing was performed using the 19.05 mm diameter, 11 deg relief angle cutting tool in order to observe the variability in the process damping coefficient. A series of three additional cutting tests were performed on the 228 Hz system with an unworn insert. Assuming a normal distribution, a two-sided 90% confidence level was computed for this small sample size. The confidence interval for the population mean was: $C = (3.2 \pm 0.15) \times 10^5$ N/m. Figure 7 illustrates the corresponding confidence region.

CONCLUSIONS

An analytical solution for machining stability while considering process damping was developed, where the process damping model was dependent on a single empirical coefficient, C . Stability testing was completed using a single degree-of freedom flexure to identify the process damping coefficient for low-speed milling of AISI 1018 steel under various conditions. It was demonstrated that a reduction in the relief angle and an increase in flank wear of the cutting edge increased the process damping coefficient.

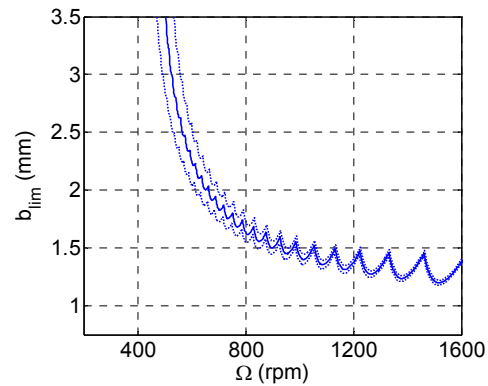


FIGURE 7. Up milling stability confidence region for 50% radial immersion, 11 deg relief angle milling tests using the 228 Hz flexure setup with an unworn cutting edge ($C = (3.2 \pm 0.15) \times 10^5$ N/m).

Process damping is particularly important for hard-to-machine materials, such as titanium, nickel super alloys, and hardened steels. In these instances, tool wear generally prohibits higher surface speeds and the use of the large stable zones available at high spindle speeds. This limits the spindle speed to low values, which decreases the material removal rate. However, by exploiting process damping, higher stable axial depths and material removal rates can be achieved.

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NANOCOINING SUB-MICROMETER FEATURES

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INTRODUCTION

The use of non-reflective surfaces in consumer and commercial applications has become increasingly popular. Whether they are used to reduce reflection on a monitor or increase the efficiency of photovoltaic solar cells, the advantages are being recognized and the demand for inexpensive coatings is high. The limitation of these nanostructured surfaces is the length of time required to produce a useable quantity, usually by deposition techniques. This paper describes an improvements made to the nanocoining procedure which increase the quality of indented areas.

The concept is to create non-reflective surfaces through a process called nanocoining. Nanocoining uses a nanostructured diamond die to imprint a structure on to a mold surface. This type of manufacturing requires a high speed actuator to be useful.

A process for creating areas of nanofeatures has been successfully developed. Areas of precisely indexed nano-indents have been created with nanofeatures exhibiting high fidelity. An elliptical tool-path has been developed to match indenter and work-piece velocity and avoid smearing the nanofeatures. However, refinements in this procedure are needed to improve the quality of each indent and thus the overall surface.

PROCESS DETAILS

There are three main elements to the nanocoining process; 1) the diamond die, 2) 2D actuator and 3) precision lathe. The die was attached to a tool holder mounted to the end of an elliptical actuator which was mounted to the vertical axis of a 4-axis diamond turning machine (DTM). The parameters of all three had to be carefully controlled to produce high quality indents.

Diamond Dies

The nanocoining process uses a diamond indenter which has a 20 x 20 μm nanostructured

area. The nanostructured area is composed of 250 x 250 x 300 nm (length x width x height) semi-pyramidal features which have either 500 or 250 nm spacing depending on the die. The 500 and 250 nm dies had 1600 and 6400 total nanofeatures respectively. An image of the nanofeatures on a 250 nm spacing diamond die can be seen in Figure 1.

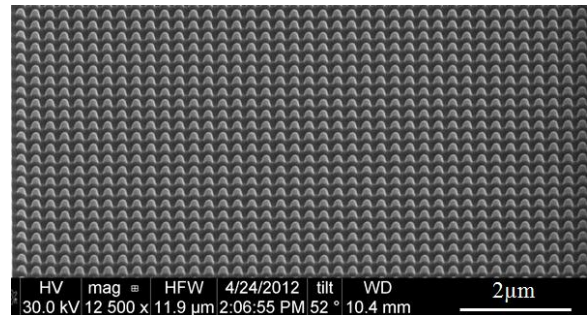


FIGURE 1. Diamond die with nanofeatures on contact face.

Two Dimensional Actuator

One of the unique aspects of the nanocoining process is matching the die and mold velocities. An actuator with a 1D tool-path (punching orthogonally into the mold surface) will cause the features to drag since the mold is constantly moving. This is what is referred to as smear. The solution to this problem was to add a second component of velocity to the tool-path in the direction of the moving mold surface. The shape of the tool-path that was chosen was an ellipse as seen in Figure 2.

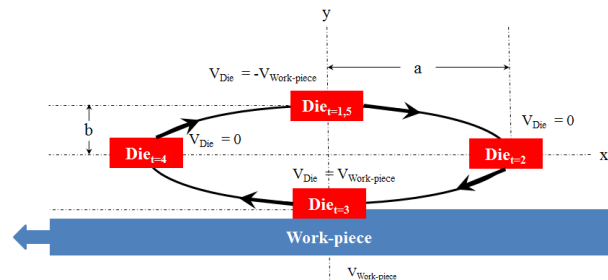


FIGURE 2. Elliptical tool-path diamond indenter must travel to avoid smear on moving work-piece.

The die is attached to a PZT driven elliptical actuator whose motion, seen in Figure 2, is a function of the indenter dimensions. The critical dimension is the horizontal amplitude, a , which is a function of the indenter length only [1]. The a dimension of the elliptical tool-path will dictate the velocity of the die in the direction of the mold. It is therefore imperative that this dimension be accurate to ensure the velocity of the indenter matches the velocity of the mold.

Nanoform 600 DTM

The Nanoform 600 DTM is a 4-axis precision lathe with 3 linear axes and 1 rotational axis in the form of an air bearing spindle. The x and z -axes are supported by hydrostatic oil bearings and driven by precision ball-screws and laser interferometers are used for position feedback on both axes. The y -axis was purchased from Moore Nanotechnology Systems and uses a hydrostatic oil bearing for the linear motion. Position feedback is achieved through an optical encoder which, when interpolation is used, can achieve step sizes of 1.1 nm. During the nanocoining process, the elliptical actuator is mounted to the y -axis and the mold is attached to the spindle by a vacuum chuck as seen in Figure 3.

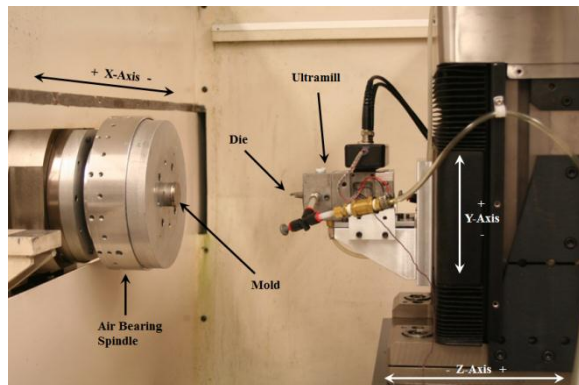


FIGURE 3. Description of nanocoining setup on Nanoform 600 DTM.

ALIGNMENT

Alignment of the die's indentation area with respect to the surface of the mold is critical to ensuring all nanofeatures are transferred from the die to the mold. An image of the different rotation angles of the die with respect to the Nanoform machine axes can be seen in Figure 4. Adjustment of the angles shown in Figure 4(b) and 4(c) is accomplished using an alignment fixture which supports the Ultramill on the Nanoform as seen in Figure 3.

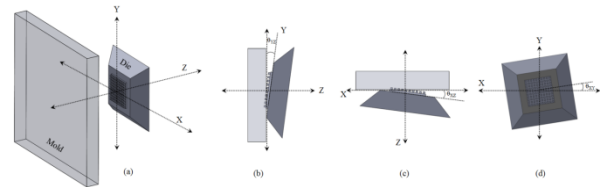


FIGURE 4. Model of die with respect to mold (a). Illustration of die misaligned in θ_{YZ} direction (b), θ_{XZ} direction (c) and θ_{XY} direction (d).

Angular deviation in these two directions can cause partial imprints. Alignment is achieved by creating indents and examining them in an optical microscope to determine if all features have been replicated. If not, adjustments are made using the alignment fixture and more indents are created. An example of this process can be seen in Figure 5.

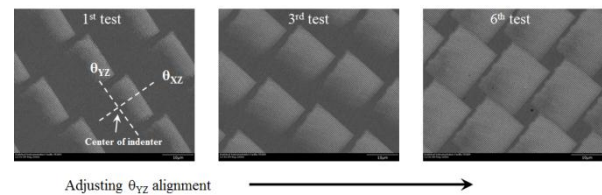


FIGURE 5. Iterations of angular adjustments. Features are completely present after 6th alignment test.

The orientation of the indenter is shown in the first image of Figure 5. Adjustment is not needed in the θ_{XZ} direction because features have been replicated completely in that direction. They have not, however, in the θ_{YZ} direction and the other images show how after several iterations of adjustment the die face is made to be parallel to the mold surface.

The θ_{XY} alignment was important for locating indents with respect to others and is called indent registration. The goal is indents created with no gaps to form a seamless area of nanofeatures. This is not possible if the die is rotated in the θ_{XY} direction and will result in gaps that can only be accounted for by overlapping. While it is not known if overlapping the features will have a negative impact on the desired property of the surface, the goal was to avoid the need to overlap. Alignment of θ_{XY} was handled by ensuring locating features on the tool-holder were square and adjustments were made by rotating the PZT stacks in the Ultramill housing. The angle shown in Figure 4(d) is adjusted by shimming the Ultramill on the alignment fixture.

NANOCOINING PROCESS

Machine code was written to move the Ultramill using the axes of the Nanoform so that large rings of indents could be created. The mold was a rotating flat so the best type of geometry to indent was a ring; future indentation experiments will use the outside surface of a rotating cylinder. Indenting on a rotating flat posed some issues because the speed of the mold will change as radial position changes (for a constant spindle speed). This was a problem because the actuation frequency of the die (and thus die velocity) remained constant and the goal is to match the velocity of the mold. The solution was to change the spindle speed and feed rate as a function of radial position using derived expressions.

INDENTATION RESULTS

Large Pitch Die Results

The nanocoin process was used to indent a 6 mm wide ring on a 1.25" diameter electroless nickel sample (stainless steel plated with nickel) at a rate of 1 KHz using the large pitch (500 nm) die. The ring started at a radius of 6 mm and ended at a radius of 12 mm resulting in an indented area of 339 mm². Based on this area and the area of the die the total number of indents was approximately 1.12 million created in 20 min. Once the part had been machined replicates were made using a UV curable material. An image of the indented part and replicates can be seen in Figure 6.

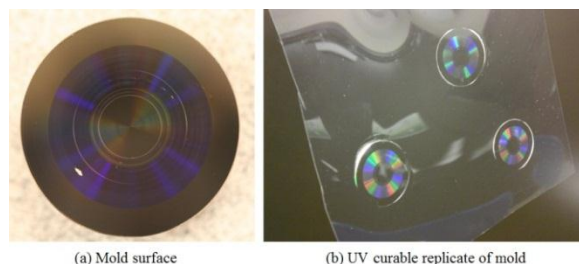


FIGURE 6. Nanocoined electroless nickel sample indented at 1 KHz (a) and replicates made from UV curable material (b).

The replicates shown in Figure 6(b) were examined in the SEM as shown in Figure 7. The large arrays and shape of features in Figure 7 resemble those found on moth eyes. Figure 7(a) also shows the overall form error of the original diamond turned mold was maintained because the die's edges did not touch.

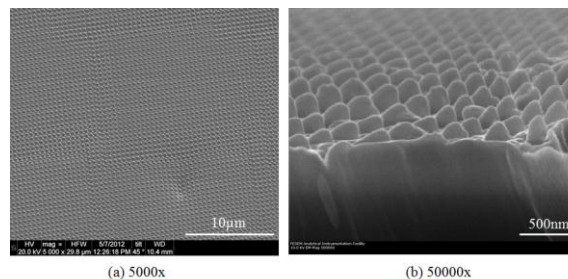


FIGURE 7. SEM images of UV curable replicate. (a) Top view, (b) side view.

This also implies the temperature control of the piezo stacks allowed the actuator to hold position. The only deformation which occurred was at the nanofeature level so no large displacement of the surface was observed. Features were replicated in the overlapped areas which were initially thought to be a problem during transfer. Figure 7(b) is a side view of a sectioned piece of the plastic replicate showing the profile of the features.

It was desired to quantify the behavior of the material around each nanofeature so an atomic force microscope (AFM) was used to quantify depth data. AFM measurements of the side of the replicate touching the mold (underside, bumps) and an inverse image representing what the mold surface would look like can be seen in Figure 8.

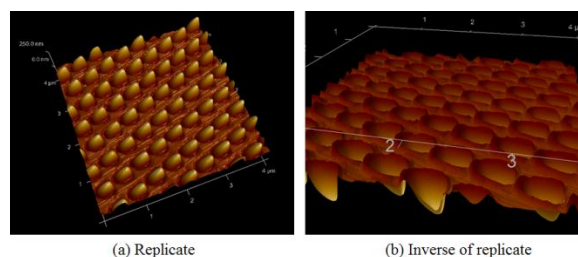


FIGURE 8. AFM measurement of UV curable replicate.

The AFM measurement in Figure 8(b) illustrates how the UV curable material formed to the surface of the mold well enough to show the displaced material around each of the nanofeature impressions. The same is true of the underside of the replicate (Figure 8(a)) which shows them as depressions around the nanofeature bumps. Data from the AFM measurement shows the depth of the depressions to be approximately 50 nm and the peaks of the nanofeatures to be 150 nm which are similar to moth's eyes. It should also be

noted that the individual features are not symmetric. The features on the die are cones with circular bases but the features in Figure 8(a) appear to have oval bases. This is caused by the smear from non-instantaneous contact time as discussed earlier. It was theorized that for the b dimension used approximately 50 nm of smear would be produced. This dragging of the features through the surface during contact would create replicates as shown.

Small Pitch Die Results

Nanocoining experiments similar to those performed with the large pitch die were then repeated using a small pitch (250 nm) die. The sub-wavelength spacing of the nanofeatures is meant for anti-reflective (AR) applications. Diffraction is avoided since the spacing is less than the wavelength of light in the visible spectrum. SEM images of the results of the AR indents can be seen in Figure 9.

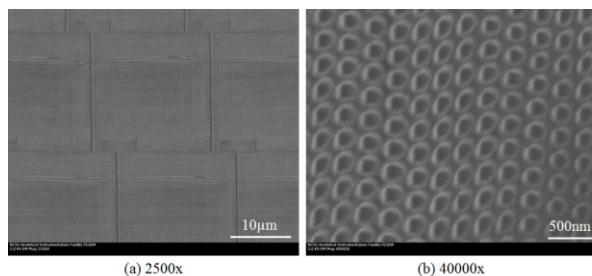


FIGURE 9. SEM images of (a) multiple AR indents and (b) individual nanofeatures.

The SEM images in Figure 9 show that the nanocoining process is capable of producing structured surfaces with sub-wavelength spacing.

NANOCOINING DRUM EXPERIMENT

The next experiment was to indent on the outside surface of a rotating drum which is more representative of the actual industrial application. To do this the Ultramill was transferred to the ASG-2500 DTM which is a three axis precision lathe (two linear, one rotational). A schematic of the Ultramill setup to indent on a drum (cylinder) is shown in Figure 10. The nanocoin drum experiments were more straight-forward than indenting on a flat since the diameter is constant implying the spindle speed and feed rate do not have to be updated.

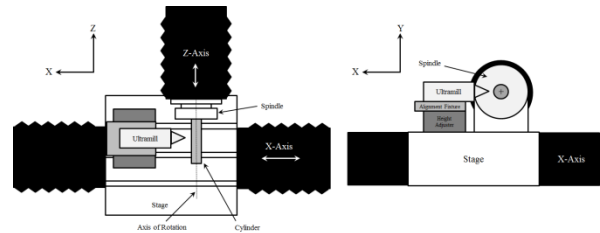


FIGURE 10. Setup on ASG-2500 DTM to nanocoin on a cylinder.

The cylindrical sample's indentation area is 25.4 mm long with a 38.1 mm diameter and plated with electroless nickel. It should take approximately 2 hours to completely cover the cylinder with nanofeatures using the 20 µm die to indent at a rate of 1 KHz. The total number of indents were approximately 7.6 million or 48.6 billion AR scale features.

CONCLUSIONS

Nanocoining is a new process that could be used to mass produce large areas of nanostructured coatings economically. A diamond die was attached to a 1 KHz actuator which had an elliptical path that matched the die's velocity with the mold velocity. Equations were developed to define the shape of the elliptical tool-path and the actuator was calibrated to produce this shape. Alignment techniques were used to make the die parallel to the mold by rotating the die about two axes of rotation to within 0.01°. The indented nickel mold and replicates created using a UV curable epoxy were examined in an SEM and AFM and the results showed the mold was able to successfully transfer features to replicates. A 250 nm sub-wavelength indenter was used to produce features which could be suitable for anti-reflective applications. An experiment has been performed to nanocoin features onto the outside surface of a rotating cylinder to mimic an industrial manufacturing process and produce a mold which could test the roll transfer production of nanofeatures.

ACKNOWLEDGEMENT

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Ellipse Shape Effect on Tool Wear in Elliptical Vibration-Assisted Machining

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INTRODUCTION

Studies have shown the benefits of applying micrometer-scale elliptical vibration motion to a tool during diamond turning (DT) [1]. Elliptical vibration-assisted machining (EVAM) causes the tool to come out of contact with the workpiece over part of each vibration cycle. Reported benefits include a decrease in machining forces, tool temperatures and wear of the diamond tool. On certain workpiece materials such as steel, nickel, or titanium, excessive chemical tool wear occurs at a rate that prohibits conventional DT of these materials. However, optical quality surfaces may be produced on these materials by incorporating EVAM. Researchers have extolled vertically aligned ellipses (ellipse major axis is perpendicular to the cutting direction) based on a phenomenon where friction on the tool rake face reverses direction and 'lifts' the chip from the workpiece, effectively reducing the average thrust force [2]. Others incorporate horizontal ellipses (ellipse major axis is parallel to cutting direction) which allow higher machining speeds [1]. However, no study could be found that directly compare vertical vs. horizontal EVAM with chemical tool wear reduction as an objective.

Tool force and temperature in EVAM has been predicted using finite element (FE) simulations [3]. This compared simulated EVAM tool temperatures as a function of frequency, ellipse shape, and cutting speed. A chemical tool wear model based on DT of steel used tool temperatures from the FE simulations to calculate wear rate. This predicted that tool wear (measured as wear per machined surface distance) using EVAM is decreased when using horizontal ellipses compared to vertical ellipses.

This paper presents results from cutting tests to validate these hypotheses drawn from FE simulation. Similar orthogonal cutting geometry used in EVAM experiments in [4] is repeated, while EVAM parameter constraints discussed in [3] are maintained for proper comparison to FE

results. This ultimately tests the capabilities of FE simulation and analytic wear modeling to predict tool wear as a function of EVAM parameters.

EVAM PARAMETRIC CONSTRAINTS

EVAM cutting conditions are set by a choice of five variables; ellipse shape (a and b , horizontal and vertical ellipse half-width amplitudes), frequency (f), depth of cut (DoC), and workpiece speed (v_{wkpc}). To study the effects of varying ellipse shape $a \times b$, the other parameters must be simultaneously varied or constrained based on resulting machining conditions such as surface finish.

The curved, cyclic motion of an EVAM tool induces a surface roughness on the part in the direction of cutting (up-feed) as well as that caused by the nose radius of the tool moving parallel to the cutting edge (cross-feed). Similar to the geometric surface finish calculation for cross-feed in conventional DT, up-feed geometric surface finish for EVAM is calculated by the following:

$$PV_{up} = \frac{b}{8} \left(\frac{v_{wkpc}}{fa} \right)^2 \text{ for } \frac{v_{wkpc}}{2\pi fa} \ll 1 \quad (1)$$

From Equation 1, one can see that vertically-aligned ellipses ($b > a$) induce a rougher surface finish than horizontally-aligned ellipses at the same v_{wkpc} . Since generating quality surface finish with EVAM is the ultimate goal, PV_{up} is kept constant for each ellipse shape tested by varying v_{wkpc} accordingly. Ellipse area (πab) is also constrained as well as DoC . DoC is made larger than the tallest vertical ellipse amplitude so that in each test condition, the ellipse center lies below the workpiece surface. Given one defined set of a , b , and v_{wkpc} parameters for one tested ellipse shape, the aforementioned constraints define the other machining conditions for comparative cutting tests.

EXPERIMENT SETUP

Experiments varying ellipse shape utilize the same experimental setup used in a previous EVAM study [4]. This used a device called the Ultramill to generate the elliptical tool tip motion. The Ultramill uses two piezo-electric stacks that actuate an alumina tool holder, shown in Figure 1. Actuation of the stacks causes elliptical motion of the tool tip that is varied with sinusoidal stack voltage and phase difference.

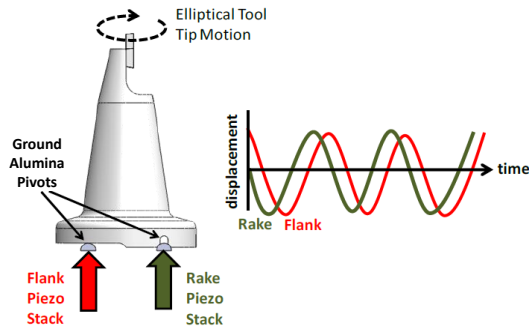


FIGURE 1. Functionality of the Ultramill and resulting elliptical tool tip motion.

Ellipse Size and Machining Parameters

To verify the ellipse shape created by the Ultramill, the tool tip motion was measured by two capacitance probes mounted at a 90° angle. These measured the displacement of a small aluminum target cube mounted to the tip of the alumina toolholder with accelerometer wax. Ellipse amplitudes a and b were mapped as a function of signal phase and peak-to-peak stack voltage amplitude, V_{p-p} . Three ellipse sizes with the same swept area were chosen and are shown in Figure 2. Workpiece speed was then chosen that satisfied the aforementioned PV_{up} constraint with values given in Table 1.

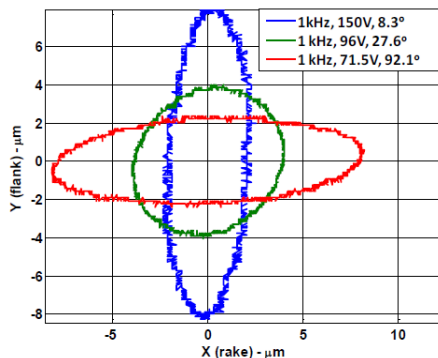


FIGURE 2. Capacitance probe measurements of Ultramill tip displacement for three ellipse shapes. Frequency, V_{p-p} stack voltage, and signal phase are given in the legend.

TABLE 1. Ultramill operating conditions and machining parameters for three cutting tests varying ellipse shape.

Ellipse shape	Horizontal	Circular	Vertical
Stack voltage	71.5 V_{p-p}	96 V_{p-p}	150 V_{p-p}
Signal phase	92.1°	27.6°	8.3°
Ellipse amplitude	8 μm x 2 μm	4 μm x 4 μm	2 μm x 8 μm
Workpiece speed	2.531 mm/sec	0.901 mm/sec	0.316 mm/sec
Total cut surface distance	5 m	5 m	5 m
Upfeed per cycle	2.5 μm	0.9 μm	0.3 μm

These parameters result in a calculated PV_{up} in Equation 1 of 24 nm for each ellipse shape, thus maintaining surface finish as a constraint. Each test used a vibration frequency of 1 kHz, a DoC of 10 μm , and total machined surface distance of 5 m.

Machining Setup

Orthogonal cutting geometry with a straight-edge tool used in [4] is repeated here. A 0.25 mm wide fin of AISI 1215 steel was mounted to a stationary post near the spindle of a 4-axis Rank Pneumo Nanoform 600 diamond turning machine. The Ultramill was mounted to the face of a Y-axis tower and cutting motion was made using the X-axis. The same single-crystal diamond tool was used for each test but a different narrow section was used for each cut as shown in Figure 3. The tool traversed the 41 mm workpiece 122 times to achieve the 5 m total cut distance for each tested ellipse.

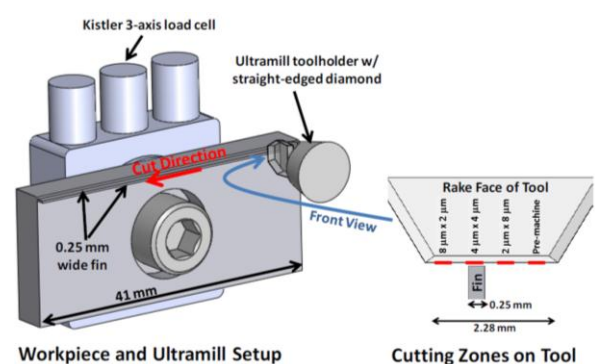


FIGURE 3. Orthogonal cutting geometry 0.25 mm wide narrow fin workpiece.

Before each cut, the fin is nominally aligned with the machining axes. A 4th zone on the tool (labeled 'Pre-machine' in Figure 3) was used prior to each test to machine the surface a depth of 25 μm to remove any oxide layer that may affect tool wear. This also created an initial surface aligned with the machining axis so DoC is consistent during initial cutting. The tool used was a single crystal synthetic diamond with 0° rake 6° clearance angles. Mineral-oil based cutting fluid dripped onto the tool to fully immerse the cutting zone. Total machining times (including non-contacting traversal of machine axes at 200 mm/min) were 2 hr 17 min for the vertical ellipse, 1 hr 55 min for the circular ellipse, and 36 minutes for the horizontal ellipse.

After each test with a particular ellipse shape, the workpiece was removed and surface finish measured using scanning white light interferometer (SWLI). Worn tool edges were measured via electron-beam induced deposition (EBID) [4,5]. This process measures the worn tool edge cross-section in a scanning electron microscope by creating a stripe of hydrocarbon contamination over the worn tool edge. This stripe is then imaged and post-processed to acquire the worn tool cross-section. Multiple cross-sections of varying wear states may then be plotted for direct comparison.

RESULTS

Post-machining surface finish results from the SWLI are shown in Figure 4. EVAM vibration marks on the surface are typically spaced by the distance called the upfeed per cycle ($ufpc = V_{wkp}/f$). These marks are visible in the horizontal and circular SWLI measurements. Cusp spacing of 2.5 μm and 0.9 μm were expected for the horizontal and circular ellipses, respectively,

and were observed on the part surface. However, the 0.3 μm spacing for vertical ellipse wasn't seen as this approaches the lateral resolution limit of the SWLI. Surface finish was similar for each test as a result of the PV_{up} constraint.

Worn tool cross-sections, measured via EBID method, are shown in Figure 5. The section of the tool used to machine with a horizontally-aligned ellipse exhibited less wear, followed by the circular ellipse section, and the vertical ellipse section showed the highest degree of wear. The wear areas, defined as the area between the worn and unworn tool edge, were 0.61 μm^2 , 1.96 μm^2 , and 2.52 μm^2 , for the horizontal, circular, and vertical ellipses, respectively.

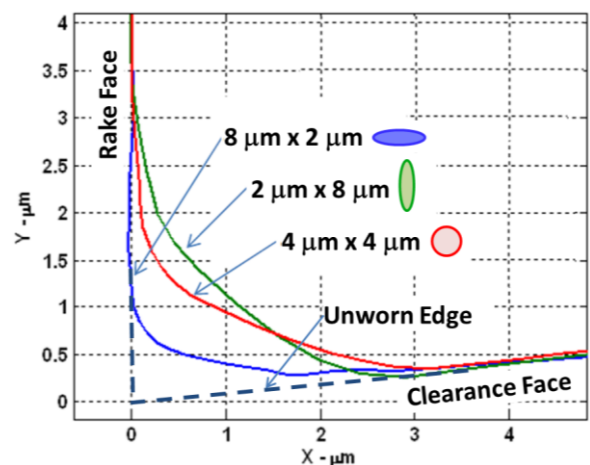


FIGURE 5. Worn tool cross-sections measured via EBID method after EVAM tests using three different ellipse shapes.

Alloy inclusions embedded in the workpiece surface are visible in the vertical ellipse SWLI

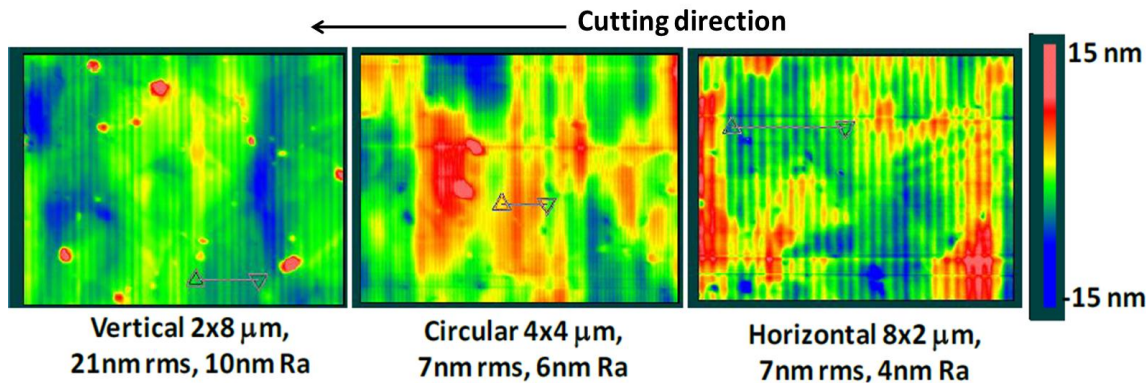


FIGURE 4. Scanning white light interferometer measurements of AISI 1215 steel after EVAM with three different ellipse shapes. Each SWLI image is 100 μm tall by 120 μm wide.

image. This same phenomenon was observed in [4] where conventional DT of the same AISI 1215 workpiece material showed these particles were dragged through the surface by the diamond tool.

DISCUSSION

Similar observations made from EVAM results in [4] were also observed here. Alloy inclusions embedded in the workpiece surface are visible in the vertical ellipse SWLI image in Figure 4. This same phenomenon was observed in [4] where conventional DT of the same AISI 1215 workpiece material showed these particles were dragged through the surface by the diamond tool.

The worn tool edges in Figure 5 showed vertical recession greater than $0.25\text{ }\mu\text{m}$ from the unworn state. This is over 100x greater than the geometric surface PV_{up} based on EVAM parameters indicating a burnishing action took place rather than cutting. Conventional DT tests on the same steel workpiece material conducted in [4] resulted in poorer surface finish than EVAM, though measured tool wear was less for the conventional DT tool. Results here follow the previous work's conclusion that excellent surface finish with EVAM is more a function of the unique cutting process rather than a reduction in tool wear.

Results from the EVAM FE study conducted in [3] showed higher peak and average tool temperatures for horizontal ellipses. Application of the thermo-chemical wear model predicted higher levels of wear per EVAM vibration cycle for horizontal ellipses. However, the surface distance traversed during each vibration ($ufpc$) is greater for the horizontal ellipse. Therefore, horizontal ellipses produce less wear over the same machined surface distance. This is principally due to the higher available workpiece speed for horizontal ellipses governed by the surface finish constraint in Equation 1, which allows shorter machining time as well. In the FE study in [4], this was described as the wear being 'spread out' over further surface distance. The lower level of wear observed after EVAM experiments here verify the FE and wear modeling predictions.

Another, simpler predictor of how wear varies with EVAM parameters is to calculate the tool-workpiece contact distance, or sliding distance. While the total cut surface distance was the

same for each EVAM ellipse tested (5 m), the total tool-workpiece contact distance is different due to the overlapping elliptical motion. During each vibration cycle, the tool contacts the workpiece over a distance greater than the $ufpc$. This distance can be calculated based on the kinematic geometry of the toolpath [6]. The calculated total sliding distance for the vertical ellipse test was 138.50 m, circular ellipse was 38.18 m, and horizontal ellipse was 20.55 m. Though temperature and wear modeling predicted less wear with the horizontal ellipse, comparing the sliding distance for varying EVAM parameters may prove to be a simpler method to predict wear trends.

ACKNOWLEDGEMENTS

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CUTTING TEMPERATURE AND VIBRATION CONTROL IN HIGH-PERFORMANCE MACHINING OF TITANIUM ALLOYS

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INTRODUCTION

Titanium alloys have recently been more widely used in the aerospace, biomedical and petroleum industries because of their good strength-to-weight ratio and superior corrosion resistance. However, it is very difficult to machine them due to their poor machinability. Diffusive tool wear is one of the dominant wear patterns for WC-Co tools in titanium machining, which is typically characterized by diffusion of the tool material into the workpiece/chip. The diffusion wear is temperature dependent; high temperatures at the tool-chip interface can favor tool wear occurring, therefore, the maximum allowable cutting temperature place an upper bound on the allowable cutting speed to achieve reasonable tool life. Consequently the maximum available spindle speed is limited, even if there are large stable zones observed in stability lobe diagrams. High temperatures are particularly problematic when the work material has low thermal conductivity. The thermal conductivity of Ti6Al4V is approximately 25 times and 10 times less than 7075 aluminum and WC-Co, respectively [1]. Such a low thermal conductivity of Ti6Al4V causes the heat to remain at the tool-chip interface, rather than being conducted away into the workpiece and cutting chips for aluminum alloys, which accelerates diffusive wear.

Chatter is a self-excited type of vibration that occurs in metal cutting if the cutting forces are relatively large with respect to the dynamics stiffness of the system. The higher cutting forces are mainly attributed to either large chip width or high strength of the work materials. Under such conditions these vibrations start and quickly grow. The cutting force becomes periodically variable, reaching considerable amplitudes. The machined surface becomes undulated and the chip thickness varies in the extreme so much that it becomes dissected.

From a machining dynamics point of view, the stability lobes tends to recommend high rotational speed to avoid the chatter frequency,

however as mentioned above the high-speed range does not work for WC-Co tools in titanium machining due to the high cutting temperature. Also the high strength of titanium alloy limits the critical chip width. Therefore, the allowed stable range for titanium alloys is very limited. Special care is needed to select the right cutting conditions so as to maintain certain productivity.

The objective of this paper is to propose a new approach to predict optimal machining conditions for finishing operation of Ti6Al4V from a machining dynamics and cutting temperature points of view.

MODELING OF CUTTING FORCES AND TEMPERATURE

The generation of mathematical models is essential for a deeper understanding of high-performance machining of titanium alloys so as to optimize it. Therefore, analytical models are needed to predict the cutting forces and temperature for high-performance machining of aerospace alloys.

Oxley [2] developed a parallel-sided shear zone theory to predict cutting forces as shown in Fig. 1. Based on Oxley's theory, Wang et al [3] proposed a hybrid cutting force model by integrating the finite element analysis and Oxley theory to estimate the cutting forces and average temperature along the shear plane and at the tool/chip interface. The concept of the hybrid model was borrowed in this study to estimate the cutting forces. Instead of depending on the FEA simulation to get the shear angle, a pure analytical approach is used in this paper to predict the cutting force based on the equilibrium condition at the tool/chip interface as that stated in [2]. Based on the assumption that the shear flow stress k_{AB} along the shear plane is constant; the shear force F_S and cutting force F_C can be estimated as:

$$F_S = k_{AB} w l = \frac{k_{AB} t_1 w}{\sin \phi} \quad (1)$$

$$F_C = F_R \cos(\lambda - \alpha) = \frac{k_{AB} t_1 w \cos(\lambda - \alpha)}{\sin \phi \cos \theta} \quad (2)$$

where w is the width of cut, l is the length of the shear plane, t_1 is the undeformed chip thickness, ϕ is the shear angle, and λ is the friction angle.

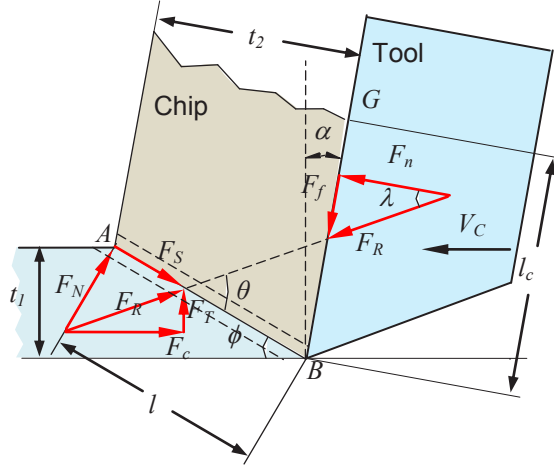


Figure 1 Cutting forces diagram based on the shear plane model

From Eq. (2), the specific cutting pressure can be estimated with the following equation:

$$K_C = \frac{k_{AB} \cos(\lambda - \alpha)}{\sin \phi \cos \theta} \quad (3)$$

In addition, the frictional force F_f at the tool chip interface and the tool-chip contact length l_c , which are two important parameters to estimate the cutting temperature in the following, can also be estimated as:

$$F_f = \frac{K_C t_1 w}{\cos(\lambda - \alpha)} \sin \lambda = \frac{k_{AB} t_1 w}{\sin \phi \cos \theta} \sin \lambda \quad (4)$$

$$l_c = \frac{t_1 \sin \theta}{\cos \lambda \sin \phi} \left\{ 1 + \frac{C_n n_1}{3[1 + 2(\pi/4 - \phi) - C_n n_1]} \right\} \quad (5)$$

where C_n is the strain rate constant, n_1 is work hardening index, and θ is the angle made by the cutting direction and resultant force.

Based on the above description, for given values of tool rake angle α , cutting speed V_C , the undeformed chip thickness t_1 and width of cut w , together with the thermal and flow stress properties of the workpiece material and the initial temperature of the work T_w (say, 25°C in all calculations), an analytical approach as given in [3] can be employed to predict the cutting forces, shear angle and the angle θ , etc.

In the prediction of the stability lobes, the cutting force coefficient is normally from empirical tests. When cutter geometries or cutting conditions are

changed, this coefficient also varies, thus tests are needed to get this coefficient again to reduce the uncertainty in the predicted stability lobes, which is costly and time consuming. With the above analytical approach, both material properties and cutter geometries are used to predict the force coefficient, which can ensure better accuracy in comparison to the conventional approach.

The temperature at the contact between the chip and tool and, in general, near to the tool tip plays an important role in the machinability of various work materials. Thus, a considerable amount of research work has been carried out to analyze temperature fields in the chip and in the tool. In this paper, a similar approach as Tlustý's finite difference computation method [4] was used with necessary augment to consider the transient heat transfer inside the cutter.

Based on the cutting mechanics shown in Fig. 2, the shear temperature along the shear plane AB T_{AB} can be estimated with the proposed cutting force model. It is assumed that the heat conducted from the chip into the surrounding air is neglected. Considering chip area $ABCD$ as shown in Fig. 2, the boundary conditions are such that along AD the chip is insulated based on the assumption that no heat conducted into the surrounding air from the chip. Along BC over the contact length l_c there is power/heat input Q_c due to chip-tool friction. The friction power is generated in the contact between the chip and the tool, and the distribution of this power follows the distribution of the normal pressure and the heat power Q_c are distributed as that in [3]. The heat spreads in the chip by conduction in the direction Y and by conduction and mass transfer (chip motion) in the direction of X . Thus, the heat balance equation for the discrete chip zone as shown in Fig. 2 is shown as the following:

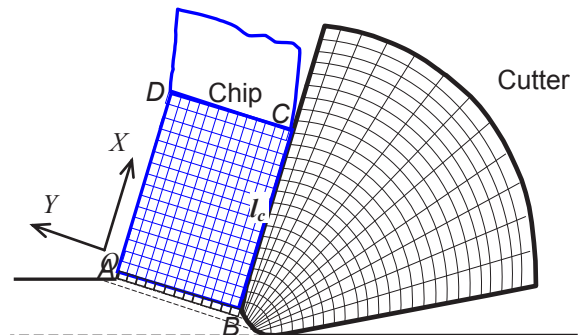


Figure 2 Finite difference formulation for temperature in the chip and cutter

$$\frac{\partial^2 T_c}{\partial x^2} + \frac{\partial^2 T_c}{\partial y^2} + \frac{Q_c}{k_c} = \left(\frac{\rho c_p}{k_c} \right) \frac{\partial T_c}{\partial t} \quad (6)$$

where T_c is the temperature field, k_c and c_p are the thermal conductivity and specific heat capacity of the work material, ρ is the density. A similar heat transfer equation (no mass transfer) exist for the tool nodal points.

As shown in Fig. 2, the basic idea of finite difference computation approach is described as follows. The whole field of the chip is divided into incremental slices in both directions. Assume that the temperature is constant over every element and changes discretely from element to element, the method of finite differences can be applied to the field. The mass transfer and heat transfer along X direction are considered. Correspondingly, only the heat conduction along Y direction are considered in the computation. Over the tool chip contact length, it obtains at its lower end heat input Q_c and the heat spreads in this slice in the direction Y by conduction, also heat is conducted away through the tool.

Based on the approach mentioned above and the input data such as cutter geometries, cutting conditions and material properties, the temperature field in the chip and cutter can be estimated, then the average temperature T_{cav} at the tool/chip contact interface can be calculated. With T_{cav} as input, the power flowing through the tool P_t can be estimated, thereafter update the power escaping through the tool for the next run and check for the relative difference in the average temperature at the tool/chip interface between two consecutive runs. The computation procedure is repeated until the temperature difference along the tool/chip is less than a threshold value, i.e., 0.005°C in this paper.

RESULTS AND DISCUSSION

At first the concept to use the analytical cutting force model to predict the cutting force coefficient was verified through experiments. A Ti6Al4V part with a convex and concave feature as shown in Fig. 3 is tested. A solid carbide end mill with a diameter of 12.7 mm was used to mill Ti6Al4V (STA) material under flood coolant and the conditions are selected as feed rate $f_z = 0.1$ mm/r-z, $V_c = 58.1$ m/min, radial depth of cut $a_e = 6$ mm and axial depth of cut $a_p = 12.7$ mm. In the test, cutting forces were measured with Kistler Dynamometer 9255B. The comparison between experimental and simulation cutting forces is demonstrated in Fig. 4. From Fig. 4, it can be seen that the predicted cutting force model can

predict the cutting forces with a good accuracy. Accordingly the cutting coefficient can be estimated with Eqs. (2) and (3).

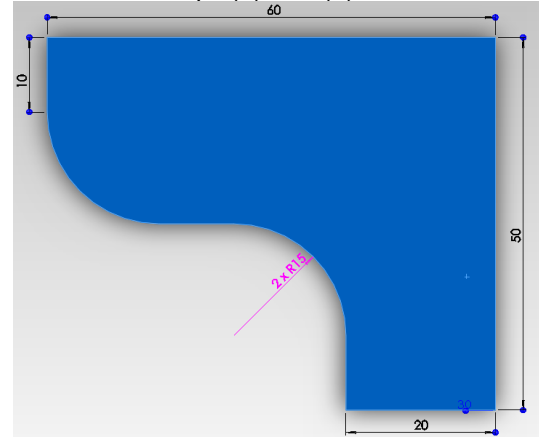


Figure 3 Feature of workpiece for cutting force testing

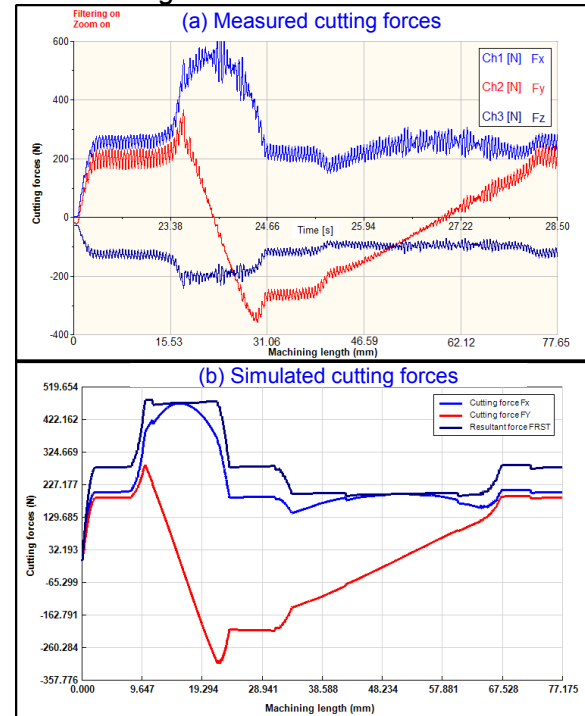


Figure 4 Comparison of experimental and simulated cutting forces

Using the cutting temperature calculation approach above, the cutting temperature distribution inside the cutter was simulated to investigate the effect of cutting speed on cutting temperature. Through the preliminary studies, it was found the cutting speed has the dominant effect on the cutting temperature, thus the main focus is on the cutting speed effect on the cutting temperature. Figure 5 shows comparison between the experimental and simulated cutting

temperature at the tool-chip interface, which was measured using the cutter/work material thermocouple method under the down milling process. The cutting conditions are selected at a feed rate of 0.1 mm/r-z and 1.0 mm radial depth of cut for a 12 mm solid carbide end mill under the dry cutting condition. It can be seen that the proposed cutting temperature calculation model predicts cutting temperature with a good accuracy. Normally the highest temperature at the rake face is about 100°C higher than the average temperature at the tool/chip contact interface. The WC-Co material loses its mechanical strength, when the maximum cutting temperature is in the range or even over 750-800°C. Thus, it is important to select cutting conditions to control the cutting temperature less than 750°C and maintain longer tool life.

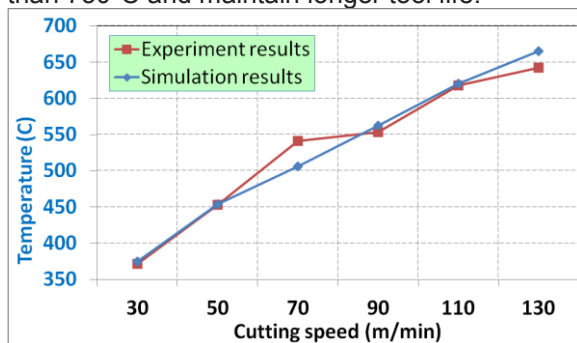


Figure 5 Comparison of experimental and simulated temperature at tool-chip interface

CASE STUDY

In this section, a case study is given to illustrate the concept on how to achieve high performance machining of Ti6Al4V by considering cutting temperature (or tool wear) and machining dynamics. A solid carbide end mill with a diameter of 19.05 mm and stick out length of 114.3 mm was used. The stability lobes of the solid carbide tool are illustrated in Fig. 6. With the increase of cutting speed, the stable cutting region tends to become wider, thus from machining dynamics point of view, higher cutting speed is preferred. However, as studied in the last section, higher cutting speed increases the cutting temperature at the tool/chip. Once the cutting temperature reaches the characteristic temperature of carbide, carbide tools cannot maintain their mechanical strength and tool wear accelerates. Therefore, due to the requirement of reasonable tool life, the cutting speed needs to be carefully controlled. Based on the stability lobes given in Fig. 6, rotational speed at 1030 rpm and 1174 rpm gives larger axial depth of cut. Under both cutting speeds, the stable

machined is achieved and the machined surface is given in Fig. 7. At 1174 rpm, the average temperature at the tool chip interface reaches 650°C, and the flank tool wear is still less than 0.1 mm after 16 passes. Thus, the optimal stable cutting conditions with a reasonable tool life are obtained.

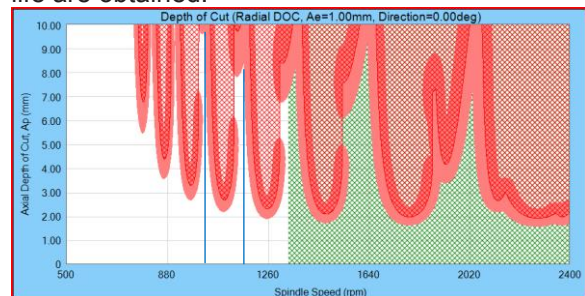


Figure 6 Stability lobes of the solid carbide tool

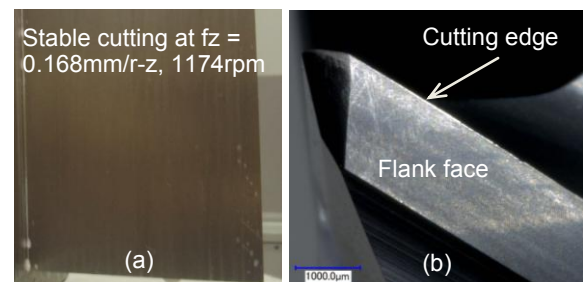


Figure 7 (a) Machined surface profile; (b) Tool flank after 16 passes.

CONCLUSION

In this paper, an analytical cutting force model is developed to predict the specific cutting pressure, followed by the temperature prediction and stability analysis based on the tap test results. Moreover experiments were performed to verify the simulation results. The results show that the proposed approach provides more stable cutting conditions as well as longer tool life using optimal cutting conditions to control the cutting temperature at the tool/chip interface under the characteristic temperature of WC-Co material.

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STUDY ON CURVED-LINE CUTTING USING A FLEXIBLE CIRCULAR SAW

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INTRODUCTIONS

The demands for carbon fiber-reinforced plastic (CFRP) for industrial applications continue to increase. CFRP products usually need to be machined (trimming process) after the curing of laminated preregs. An abrasive water jet or endmill cutting is often used for the trimming process; however, these methods present problems regarding machining efficiency and cost [1][2].

Here we propose a flexible circular saw as the core of a new process for CFRP plate cutting. A normal circular saw has several cutting edges, and it rotates at high rotational speed [3], providing high-speed cutting (feed rate 3000 mm/min) that is about three times higher than the speed of other machining methods [4]. However, a circular saw is designed for straight line cutting [5]. It is thus difficult to cut a continuous curved line without interference between the saw body and the machined surface. One way to avoid this interference would be to bend the circular saw to fit the target curved line. A flexible circular saw could be deflected, like a bowl-shape. This can be realized by acting circular forced displacement while the center is held. A cross-section of a flexible saw's body making a circular arc is shown in Fig. 1. Curved lines could be cut without interference by the bowl-like deflection. In our previous study, it was confirmed analytically that the flexible circular saw can cut constant curvature line without interference by using FEM [6].

In this paper, actual curved line cutting tests for CFRP plate were conducted. In addition, a new control method of the developed machine tool for a flexible circular saw that can cut continuous curved line was proposed.

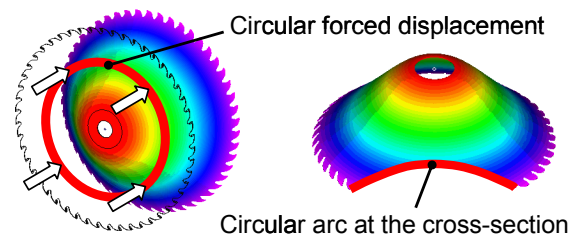


FIGURE 1. Schematic of flexible circular saw to cut curved line.

CUTTING TEST FOR CFRP PLATE USING FLEXIBLE CIRCULAR SAW

In the previous study, it was shown that the cross-sectional shape of the deflected saw body can be approximated as a circular arc by FEM analysis [6]. However it was not applied to the actual cutting. In order to confirm the feasibility, cutting test for CFRP plate was conducted on a 4-axis control NC machine tool which had been specially developed for flexible circular saw (Fig. 2). The C-axis is a swiveling axis for changing the sawing direction. The U-axis is a linear axis for changing the deflection of the saw body. A target shape of the test was a circular arc with R5000 mm. The cutting conditions are shown in Table 1. The workpiece was a CFRP plate for aircraft. The fiber direction was alternately laminated at 0° and 90°.

To confirm the cutting accuracy, we measured the machined outline with a three-coordinate measuring machine. The results are shown in Fig. 3. The circular arc of the curvature radius R5000 mm is also drawn in the figure. On the top face of the workpiece, it is clear that the measurement values agreed well with the circular arc. However, on the bottom face of the workpiece, the measurement values are always smaller than that of the top face. This means that cross-section of the machined surface is inclined. The maximum inclination angle was

1.79° The mechanism why the machined surface inclines is as follows. Cutting edges are located on the circle at the outer circumference of the saw body even if the saw body is deflected like bowl-shape. The machined surface is generated as the enveloping surface of the movement of this circle. The movement of the saw is the combination of the linear motion and the swing motion at the curved line cutting. On the swing motion, distance between the swing center and the circle is not constant, as the location of the saw is lower, distance from the swing center becomes small. Then the machined surface inclines.

A surface roughness was also measured. In the measurement, we measured along laminating direction and feed direction. The result is shown in Fig. 4. From this figure, it is clear that the surface roughness on the laminating direction was larger than that on the feed direction. This is because that the cutting property is different depending on the fiber orientation. Maximum surface roughness is 5.3 μmRa . This value is small enough for the rough cutting of CFRP. In addition, when the machined surface was observed by a microscope, pull-out of the fibers, or signs of melting of the matrix resin was not observed on the surface. However, at the bottom face of the machined workpiece, a few delaminations, size of which was less than 0.5mm, were observed as shown in Fig. 5.

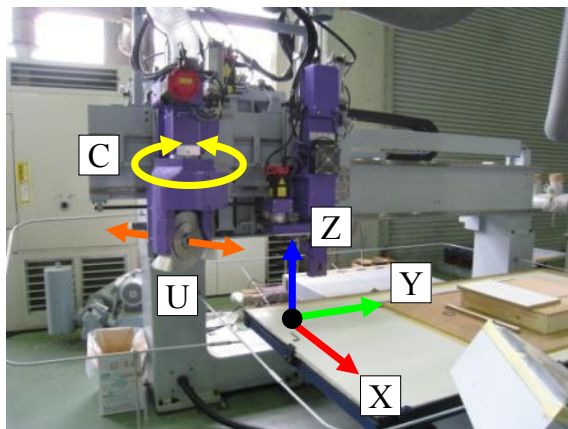


FIGURE 2. 4-axis control NC machine tool for a flexible circular saw.

TABLE 1. Cutting conditions.

Material		CFRP
Material thickness	mm	10
Cutting length	mm	480
Feed rate	mm/min	3000
Spindle speed	min^{-1}	3000
Cutting direction		Down cut
Saw diameter	mm	305
Saw thickness	mm	1.5
Tooth width	mm	3.5

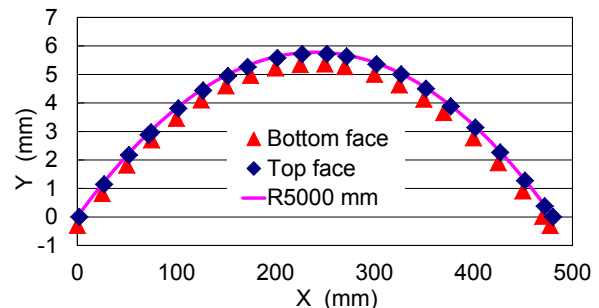


FIGURE 3. Comparison of shape between measurement and circular arc.

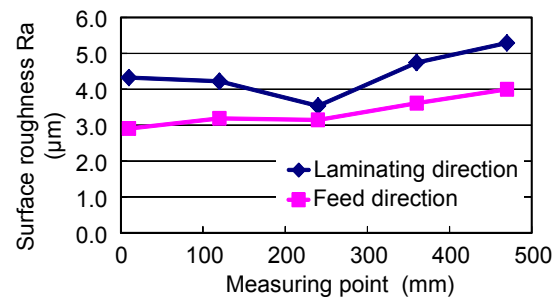


FIGURE 4. Surface roughness of CFRP plate.

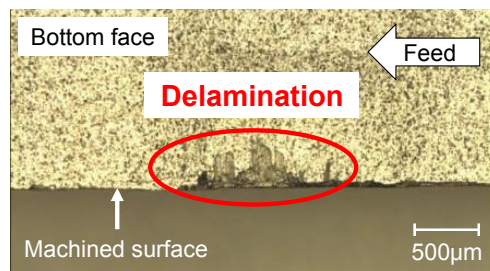


FIGURE 5. Delamination of CFRP plate.

CONTINUOUS CURVED LINES CUTTING USING FLEXIBLE CIRCULAR SAW

It is difficult to cut continuous curves combining concave and convex shapes using flexible circular saw because it causes interference between the machined surface and the saw body. Therefore, in order to apply for the

practical cutting, a new method that can cut continuous curves using the saw is proposed.

A circular saw has clearances between a saw body and a cutting edge. The width of the cutting edge is larger than the width of the saw body in order to avoid a friction between the machined surface and the saw body as shown in Fig. 6(a). The proposed method utilizes this clearance. The saw body can deflect within this clearance without the interference while cutting as shown in Fig. 6(b). Then, the curves can be cut continuously by changing the deflection of the saw body and the direction of the saw successively as shown in Fig. 6(c). Specific procedures of the method are shown as Fig. 7:

- (a) Find the next location of the preceding cutting edge along the aimed curve where the distance from the current point is incremental step ΔL .
- (b) The circular arc of the saw body should be tangential to the aimed curve at the next location of the preceding cutting edge. And the deflection of the saw body should be defined so as not to interfere with both sides of the machined surface A and B. In addition, next location of the following cutting edge should be on the aimed curves.
- (c) Then, the saw moves forward with linear interpolation of the change of the deflection and the inclination of the saw body.
- (d) Above procedure is repeated.

If target continuous curves fulfill C1 continuity, this "continuous curved lines cutting method" theoretically can be applied for any continuous curves (ex. NURBS curves, etc.). However, curvature of the target shape is limited by the diameter and the clearance of the circular saw. The length of the cross-section of the saw body becomes long as the diameter of the saw becomes large. The flexion point on the curve between the preceding cutting edge and the following cutting edge should be only one. Because the deflection shape of the flexible circular saw is the shape of a first-order mode (bowl-like shape). Thus, the applicable wave length of the target curves becomes long as the length of the cross-section of the saw becomes long. In addition, the relationship between the width of the cutting edge and the thickness of the saw body affect the minimum radius of curvature of the target curves. When the width of the cutting edge is larger or the thickness of the saw body is smaller, the clearance is larger. Therefore, the minimum radius of curvature of the target is smaller. In reverse situation, the

minimum radius is larger. Therefore, size of the saw should be selected properly.

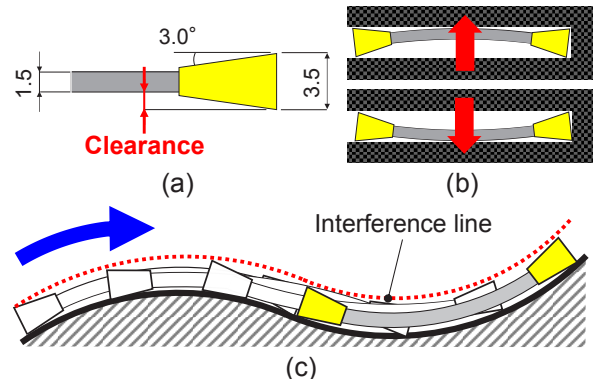


FIGURE 6. Continuous curved lines cutting method.

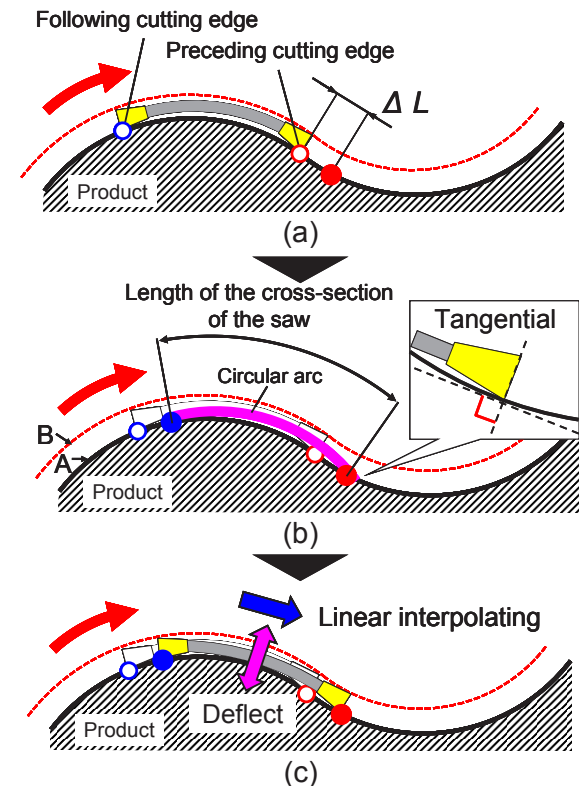


FIGURE 7. Procedure of continuous curved lines cutting method.

In order to confirm the feasibility of this method, cutting test was conducted. Target shape of the test was the combination of tangentially connected two circular arcs with radius R5000 mm. A workpiece was a blockboard whose thickness was 10 mm. The others cutting conditions were the same as in Table 1. The cutting test was conducted using two NC programs that had been divided the curves into 20 segments and 40 segments.

To confirm the cutting accuracy, we measured the machined outline with a three-coordinate measuring machine. The results of the measurements are shown Fig. 8. It is clear that there are machining errors between the measurement and the target shape. The maximum error is 4.12 mm on the condition of 20 segments. Furthermore the burns of the workpiece caused by the interference of the saw body and the machined surface were observed at the cutting length 150 mm and 500 mm. This error was caused by a motion error of the machine tool. When the actual deflection of the saw body was measured with dial gauge, the actual deflection was smaller than the target deflection. Several link mechanisms are used to give a circular forced displacement on the circular saw. The link mechanism has small gap for smooth action. Therefore, the actual displacement becomes small by the accumulated errors of each link mechanism. The infection of the small-deflection is shown in Fig. 9. When X is between 100mm and 350 mm as shown in Fig. 9(a), the actual deflection was smaller than the target deflection. The location of the cutting edge departed from the product. When X is between 350mm and 700mm as shown in Fig. 9(b), The location of the cutting edge cut into the product.

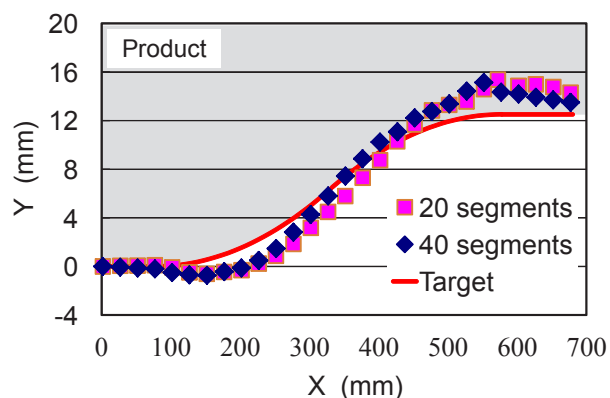


FIGURE 8. Comparison between measurement and target shape.

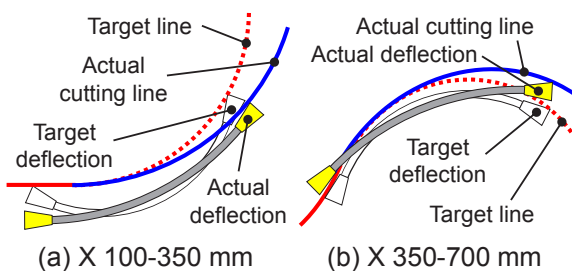


FIGURE 9. Infection of the small-deflection.

CONCLUSIONS

A flexible circular saw is a novel machining method that can cut the curved line for the trimming of CFRP plate. Then, the feasibility of the saw was researched. A new control method that can be applied for all continuous curves with a flexible circular saw was also proposed. The results are summarized as shown below.

- 1) Cutting test using a flexible circular saw for a CFRP plate was conducted. As a result, it is found that the saw can cut the CFRP with high speed and good accuracy.
- 2) A new method that can cut all continuous curves using flexible circular saw was proposed.
- 3) In order to confirm the feasibility of the continuous curved line cutting method, cutting test was conducted. As a result, the continuous curves could be cut using the method. However, there was difference between the measurement and the target shape because of the motion error of the machine tool.

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REAL-TIME FEED OPTIMIZATION FOR SPLINE TOOLPATHS EMPLOYING UNINTERRUPTED ACCELERATION PROFILING

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INTRODUCTION

This paper presents a new real-time algorithm for generating time-optimized feedrate profiles that are suitable for contouring along spline toolpaths. The solution of this problem has interested the robotics and machine tools community for a long time. Some initial solutions proposed have been simple to implement, yet had the shortcoming of producing discontinuous acceleration profiles which are detrimental to the performance of precision machinery [1][2].

Other solutions have been proposed which rely on elaborate optimization techniques to compute the trajectory that yields the shortest cycle time, while preserving the continuity of acceleration [3][4][5][6]. As a compromise, practical solutions have also been developed, which modulate the feed profile in the form of piecewise constant steps with smooth jerk limited transitions in between the adjacent segments [7][8]. In spite of the simplicity and usefulness of this approach, one problem associated with it is that gradual accelerations spanning multiple segments cannot be achieved without interrupting the tangential acceleration profile to go through zero near each segment crossing, thereby causing unnecessary jerky motion and increasing the overall cycle time. This issue has been identified in the CNC community as the so-called 'zero-acceleration' problem.

The main feature of the proposed algorithm is its ability to modulate the tangential acceleration across a large number of trajectory segments (i.e., NC blocks) without this problem. The proposed technique, therefore, can provide substantial reduction in the cycle time by reducing the distance required to accelerate to the desired feed value when traversing through multiple segments. This also helps in improving the motion smoothness. As the resulting trajectory is void of unnecessary jerk transients, it is easier to track by the servo control system.

OBJECTIVE AND CONSTRAINTS

The objective is to minimize the total motion duration. Defining the path parameter as a function of time $s(t)$, and its derivative (the feedrate) $v = ds/dt$, the objective can be expressed as:

$$J = \int_0^T dt = \int_0^S \frac{ds}{v} \quad (1)$$

Clearly, maximizing the feed throughout the toolpath minimizes the total travel time.

The constraints considered are the limits on the tangential acceleration and jerk (A, J), and a feed limit f_k designated for each spline segment. The feed limits are determined by considering the toolpath geometric derivatives and actuator velocity, acceleration, and jerk limits [7][8]. Limits coming from the machining process due to tool deflections and cutting forces can also be considered [9]. Hence:

$$\left. \begin{aligned} |a| &= |dv/dt| = |d^2s/dt^2| \leq A \\ |j| &= |d^2v/dt^2| = |d^3s/dt^3| \leq J \\ 0 \leq v &\leq f_k, \quad k: \text{toolpath segment number} \end{aligned} \right\} \quad (2)$$

PROPOSED SOLUTION

The proposed solution generates the time-optimized feed profile by alternating between two basic feed types shown in Fig.1. Both profiles end at zero velocity and acceleration, and are formulated according to the standard S-curve shape by employing linear acceleration transients [9]. In their computation, the tangential acceleration and jerk limits in Eq. (2) are used.

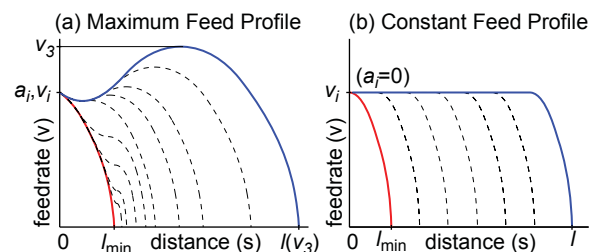


FIGURE 1. Basic profiles used in the algorithm.

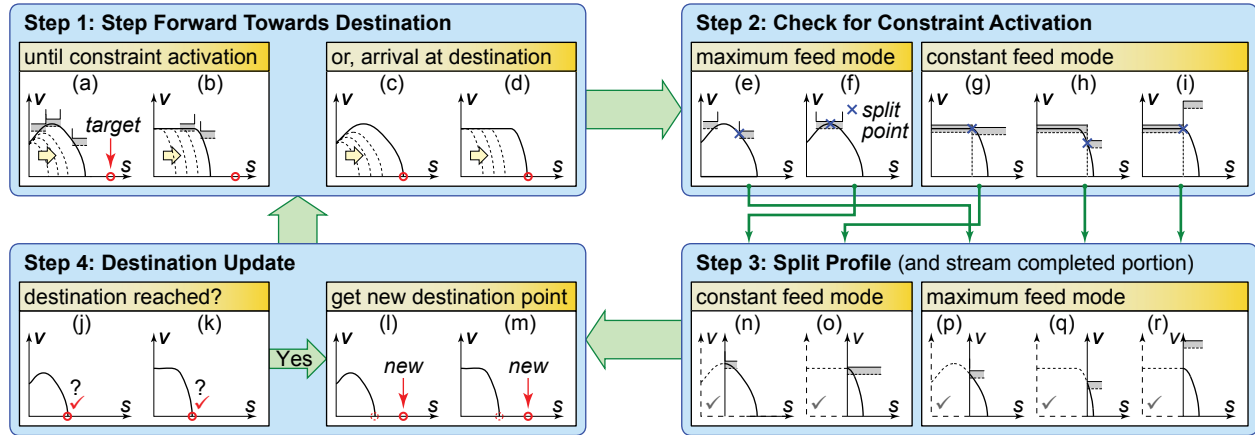


FIGURE 2. Steps of the proposed algorithm.

Maximum Feed Profile

This profile can be generated with arbitrary initial acceleration ($|a_i| \leq A$) and velocity (v_i) boundary conditions. The initial conditions are inherited from a state on an earlier planned feasible trajectory (as explained in Step 3 of the algorithm), thus guaranteeing that all tested $l > l_{\min}$ will also yield profiles with kinematic feasibility. As the travel distance (l) increases, so does the maximum feed value (v_3), which guarantees that for sufficiently large l , the feed profile approaches the vicinity of at least one feed limit (i.e., $f_k - \delta v \leq v(s) \leq f_k$) and results in constraint activation.

Constant Feed Profile

This profile is generated with zero initial acceleration ($a_i = 0$) and a given initial feed (v_i). The commanded travel distance determines the duration of constant feed. This profile is invoked when constraint activation is encountered at the peak feed value. Again, all tested $l > l_{\min}$ cases yield kinematic feasibility.

Overall Algorithm

Progression of the proposed feed optimization is shown in Fig. 2. The algorithm starts by generating a maximum feed profile with zero initial conditions and travel distance. Upon reading the path distance and feed limit for the first segment, the following cycle starts to iterate:

Step 1: The travel length is increased using bisection either until constraint activation or arrival at the current destination.

Step 2: If any of the feed limit constraints are active, the one closest to the end of the profile determines the splitting point. This point coincides with a toolpath segment boundary or the maximum feed point (where v_3 occurs).

Step 3: If any feed constraints are active, the profile is split into two at the designated point. The left-hand side is complete and ready to be streamed to the real-time interpolator. The right-hand side constitutes the initial guess for the remainder of the solution.

Step 4: If the current destination is reached, a new toolpath segment with a new feed limit and destination point is read from the input stream.

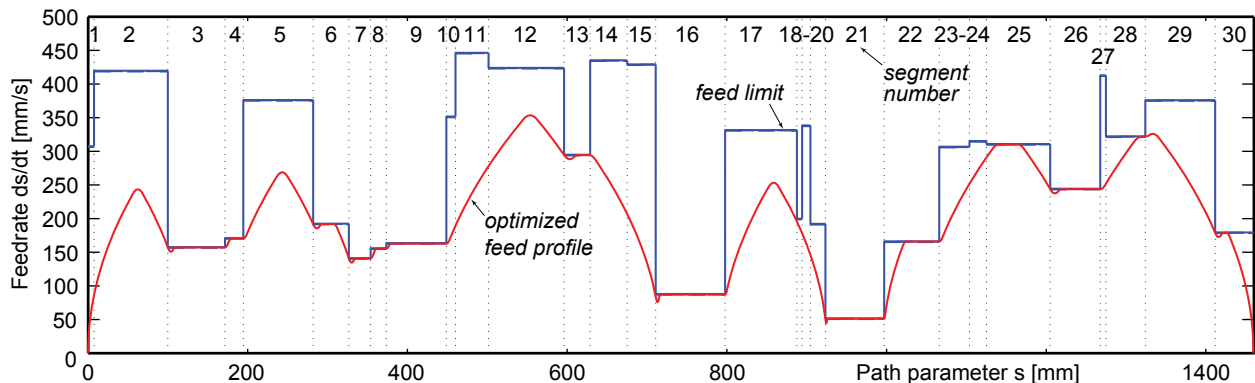


FIGURE 3. Sample run of the algorithm for a 30-segment toolpath ($A = 0.5 \text{ m/s}^2$, $J = 20 \text{ m/s}^3$).

IMPLEMENTATION RESULTS

The algorithm has been implemented in Matlab® and its stability has been validated through a rigorous state analysis involving all decision variables and kinematic possibilities. The code has been tested for several sets of toolpaths comprising of 100,000 randomly generated feed segments and different acceleration and jerk limits. Fig. 3 presents a sample result, generated for only 30 segments for visualization purposes. It can be seen that the feed limits are well utilized, and that acceleration transitions occur seamlessly across multiple segments without interruption or unnecessary jerk. The average block processing duration was measured to be ~20 ms for the Matlab code running on a 2.26 GHz Core Duo CPU with Windows 7-64.

INTEGRATION WITH SPLINE PROFILING

The algorithm is currently being integrated with a real-time spline interpolator for further testing in surface and contour machining experiments.

The algorithm is shown in Fig. 4. First, the toolpath shape is parameterized by reading data points from a continuous stream of coordinates and fitting cubic B-splines on the fly [10]. Toolpath segments are chopped up into sub-segments, in order to allow finer calculation of local feed limits. The arc lengths are integrated using Simpson's method, and simultaneously geometric derivatives of axis position $\mathbf{r} = [x, y, z]$ with respect to the arc parameter s are computed ($\mathbf{r}_s, \mathbf{r}_{ss}, \mathbf{r}_{sss}$). The relationship between s and the corresponding spline parameter u is computed as the 'feed correction polynomial' [11], using data generated during arc length integration. This allows the toolpath to be interpolated without unwanted feed fluctuations. The feed limits (f_k) are computed for each sub-segment by considering the maximum allowable velocity, acceleration, and jerk magnitudes ($\dot{r}_{\max}, \ddot{r}_{\max}, \dddot{r}_{\max}$), and utilizing the geometric derivatives and upper bounds for the expected magnitudes of A and J , in the following constraints:

$$\left. \begin{aligned} |\mathbf{r}_s f_k| &\leq \dot{r}_{\max} \\ |\mathbf{r}_{ss} f_k^2 + \mathbf{r}_s A| &\leq \ddot{r}_{\max} \\ |\mathbf{r}_{sss} f_k^3 + 3\mathbf{r}_{ss} A f_k + \mathbf{r}_s J| &\leq \dddot{r}_{\max} \end{aligned} \right\} \quad (3)$$

The computed feed limits, along with other toolpath information, are processed using the proposed feed optimization technique. The

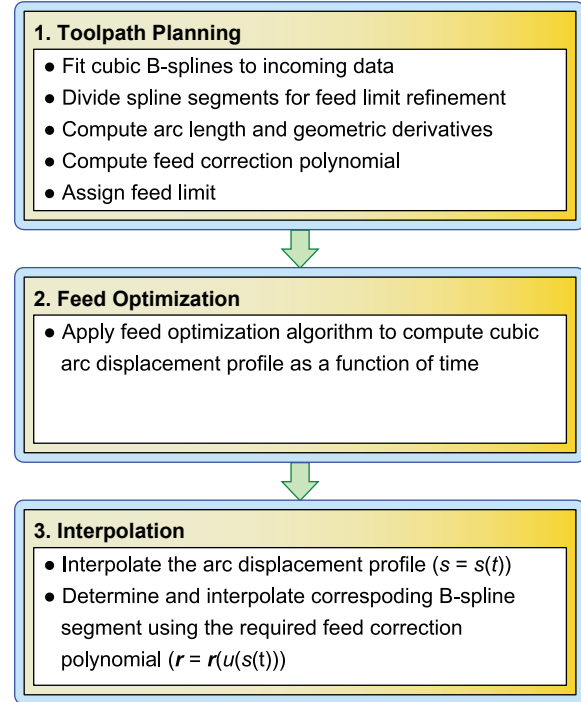


FIGURE 4. Overall trajectory generation scheme.

optimized arc displacement profiles are then interpolated as a function of time, and corresponding position values on the toolpath are resolved using the fitted B-spline data.

A sample implementation result is presented in Fig. 5. The feed limits computed for each segment are shown with dashed lines in the phase plane plot. As can be seen, the optimized feed profile closely approaches the limits while also respecting the tangential acceleration and jerk bounds, which were set as $A = 1.25 \text{ m/s}^2$, $J = 50 \text{ m/s}^3$. The axis velocity, acceleration, and jerk graphs indicate that these profiles never exceed their respective limits (which are designated with the dashed lines). The kinematic capabilities of each axis are reasonably well utilized.

The fact that the actual velocity, acceleration, and jerk profiles do not necessarily hit their limits originates from the conservative calculation of f_k from Eq. (3). This assumes that the direction and magnitude of the tangential acceleration and jerk is not known in advance. Therefore, the worst-possible case is considered in order to guarantee that these constraints always hold. While the magnitude of J is assumed to be given, the maximum tangential acceleration A

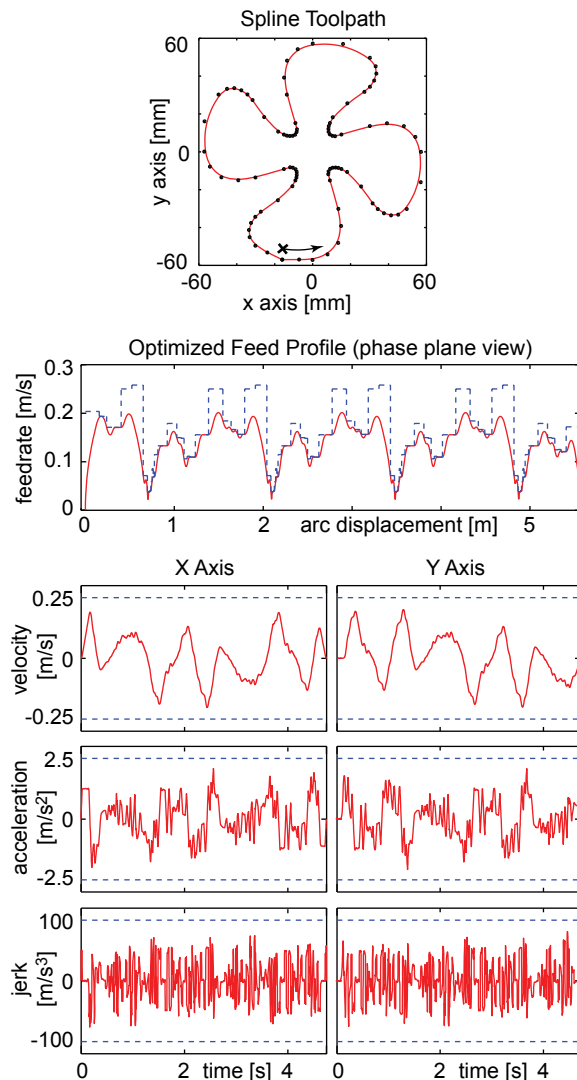


FIGURE 5. Implementation result.

is computed by considering the envelope of possible velocity and acceleration values that follow a standard S-curve [9], thereby allowing less conservative feed limits to be estimated for the intervals $[0, A^2/2J]$ and $[V - A^2/2J, V]$.

CONCLUSIONS

This paper has presented a practical and effective way to optimize the feed profile for traveling along spline toolpaths. Piecewise constant feed limits are computed as a function of the path parameter. The feed profile is modulated to enable uninterrupted acceleration across multiple segments, while approaching the feed limits as closely as possible. The methodology is simple and efficient, and can be implemented inside the block processing sequence of a CNC system.

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DRIVING MODE CONTROL OF A MULTILAYERED THIN ELECTROSTATIC ACTUATOR SUPPORTED BY LUBRICATING OIL FOR ULTRA-PRECISION POSITIONING

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INTRODUCTION

Fine stages are widely used in microscope systems, optical systems, and manufacturing systems. Electrostatic actuators have some advantages in terms of their low heat production and the large number of available materials. They are also able to transmit power without mechanical contact and do not require hinges. In addition, the multilayered structure can increase the produced thrust force.

Several types of electrostatic actuators have been studied which are supported by ball guides and enable a longer working range of motion that exceeds 200 μm . The ball guides are effective at increasing the advantages of electrostatic actuators [1, 2]. However, they require complex fabrication processes for the guide structure and these mechanisms retain numerous beads between the mover and stator layers which are used to maintain the gap.

As the solution, an electrostatic actuator supported by only lubricating oil has been proposed [3]. In the actuator, the motion characteristics depend on the driving signal since the frictional condition is changed by the driving signal. It is shown that a suitable pulse driving signal provides low friction, a long working range and high positioning accuracy.

As an additional requirement, fine positioning with continuous large holding force is also required. In this paper, the friction force is continuously utilized as the holding force. The electrostatic actuator has the ability to change the friction force by using the suitable driving signals. This paper describes two driving modes provided by the different driving signals and the ultra-precision positioning based on the combination of the driving modes.

ACTUATOR STRUCTURE

Figure 1 shows the driving method for the bi-directional motion of a multilayer electrostatic actuator supported by only lubricating oil. The actuator has two mover layers and three stator

layers. Each layer is covered with ethylene tetrafluoroethylene (ETFE) flat film as an isolation film. Only the lubricating oil is inserted between the layers.

The mover layers are grounded. When the absolute value of voltage V_1 is not equal to zero and voltage V_2 is equal to zero, the mover produces a thrust force toward the left due to the electrostatic attractive force between the electrodes (see Fig. 1(a)). As shown in Fig. 1(b), when the absolute value of voltage V_2 is not equal to zero and voltage V_1 is equal to zero, the mover produces a thrust force toward the right.

When the absolute voltages of V_1 and V_2 are the same, the mover does not produce a thrust force. However the friction force between them increases and acts as the holding force.

Figure 2 shows the overall view of the experimental setup. In the figure, a prototype of the multilayer electrostatic actuator is located to the left side. The actuator comprises a mover having two electrode layers and a stator having three electrode layers. Each electrode has 26 slits. As a lubricating oil, the silicone oil of which viscosity is 10 mPa.s is inserted between electrode layers and used to maintain the gap between them. The prototype is designed to have the working range of 0.5mm and driven by

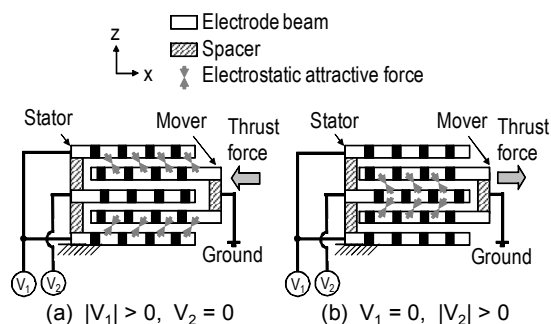


FIGURE 1. Driving method for the bi-directional motion of a multilayer electrostatic actuator supported by only lubricating oil.

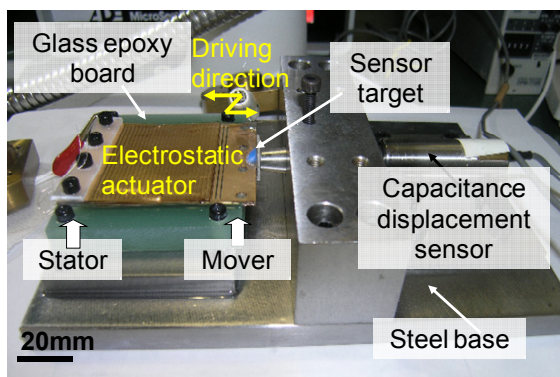


FIGURE 2. Overview of the displacement measurement setup for the electrostatic actuator.

using two high voltage power amplifiers. For measuring the displacement of the mover, a capacitance displacement sensor is used. The motion range for evaluating the actuator characteristics is limited by the displacement sensor of which measurement range is 50 μm . When the force characteristics are measured, the displacement sensor is replaced with a load cell.

EFFECT OF DRIVING SIGNAL

In multilayer electrostatic actuators supported by only lubricating oils, the attractive electrostatic force between electrode layers greatly influences the gap between them and changes the friction force between them. Thus the electrostatic actuator has the ability to change the friction force by using the suitable driving signals. Based on the ability, the electrostatic actuator has two driving modes. One of the driving modes is for driving the mover under the low frictional effect in a wide working range (referred to as the wide driving mode) [3]. In the other mode, the normal control signals with a constant signal are used for a short working range motion. The signals lead to the large friction force between the electrode layers and shorten the working range of the mover. However the limited working range is not zero and the friction force can be used as a holding force.

Holding Force

Figure 3 shows the relationship between the applied voltages and the holding forces for the actuator resulting in the friction forces. In the experiments, the voltage of V_1 was set to equal to the voltage of V_2 . The force was measured

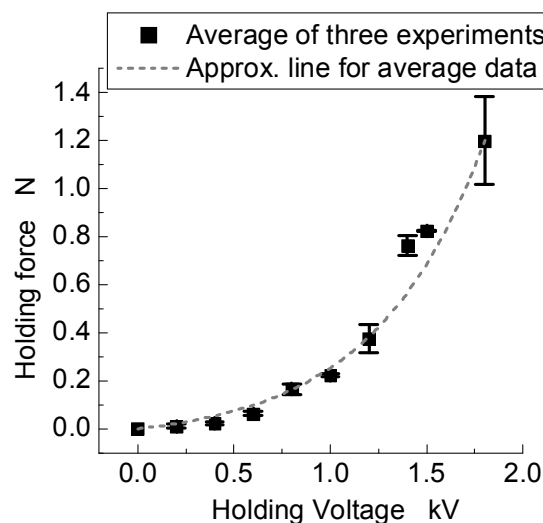


FIGURE 3. Relationship between the applied holding voltages and the holding forces.

with a load cell. The symbols refer to the average data of three similar experiments. The curve represents the average approximate line calculated using the least-squares method. From this figure, it can be concluded that the holding force increases with the increase in the applied voltage due to the increased attractive forces between them. The applied voltage consequently generates a large holding force. It also can be seen that for a 1 kV applied voltage signal, the actuator has a holding force of 0.205 N. Additionally, the holding force by the holding voltage higher than 1.5 kV is expected to be higher than the thrust force produced by the wide driving mode signal of which amplitude equals to the holding voltage.

Fine Driving mode

Fine positioning with continuous large holding force is desired, for example, in microscope systems and optical systems. For the fine positioning, the combination of the normal control signal and the holding signal of a constant voltage is used. This driving mode is referred to as the fine driving mode.

Figure 4 shows the experimental open-loop displacement response of the actuator to a normal driving signal with a 1kV holding voltage. The holding voltage of 1kV generates a static force of 0.205 N. In the figure, the mover reciprocates in the range wider than 120nm. The result indicates that the actuator can finely and precisely adjust the mover position with the holding force.

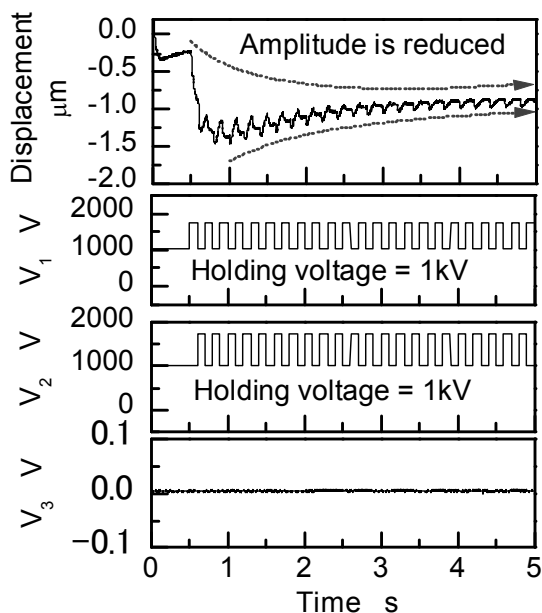


FIGURE 4. Experimental open-loop displacement response of the actuator to a normal driving signal with a 1kV holding voltage.

Effect of Fine Driving Mode Signal on Motion in Wide Driving Mode

The combination of the two driving modes (referred as to dual driving mode) has the potential to achieve the point-to-point (PTP) positioning in the wider working range and the ultra-precision adjustment with the continuous holding force near the reference position. In the PTP positioning, the switching characteristics between the modes greatly influence the response of the control system with the actuator. The actuator is in bad condition for the wide driving mode right after the fine driving mode. There is no gap between the electrode layers. In addition, the fine driving mode signal can cause charging between the electrode layers since they have the large DC component.

Figure 5 shows the open-loop displacement response to the monopulse signal for the wide driving mode after applying a 1 kV holding signal. This indicates that the monopulse signal in [3] does not produce the significant displacement under the bad condition.

Figure 6 shows the open-loop displacement response to the different monopulse signal of which the sign is different from that of the holding signal. The initial peak to peak value of the displacement is about 10μm and the p-p

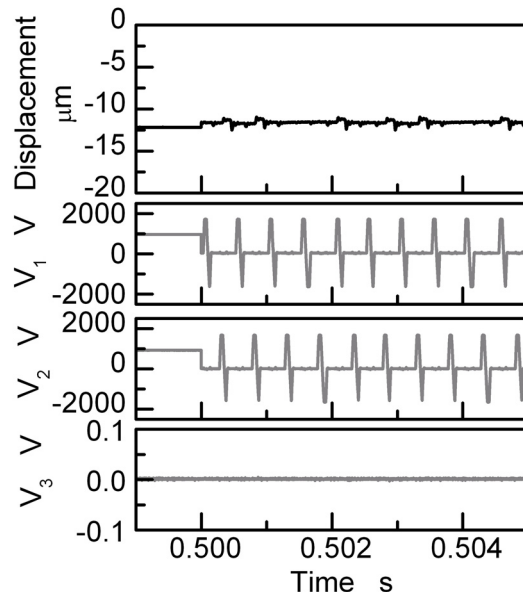


FIGURE 5 Experimental open-loop displacement response to the monopulse signal for the wide driving mode after applying a 1 kV holding signal

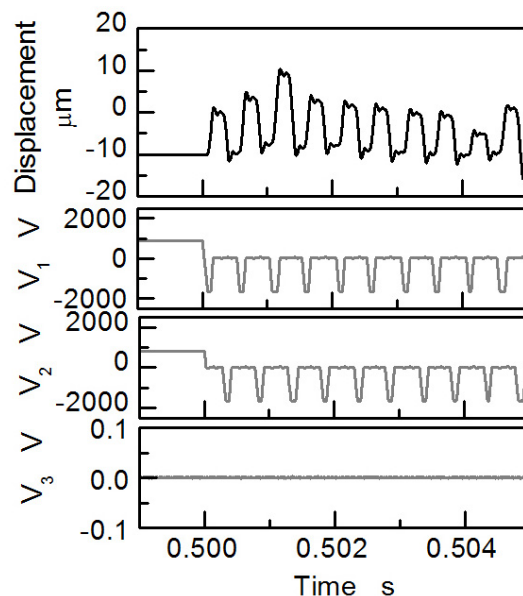


FIGURE 6 Experimental open-loop displacement response to the different monopulse signal of which the sign is different from that of the holding signal after applying a 1 kV holding signal

value does not decrease very well. This result indicates that this monopulse signal is useful as the improved wide driving mode signal for the combination of the fine driving mode and the wide driving mode.

DUAL DRIVING MODE CONTROL FOR ULTRA-PRECISION POSITIONING

Using the improved monopulse signal as the wide driving mode signal, PTP positioning with the dual driving mode was carried out. Figure 7 shows the block diagram of the control system for the PTP positioning. The control system includes the PID controller, a linearizer, and a driving signal unit. The linearizer is used to cancel the nonlinear characteristics between the input voltage and the thrust force. The driving signal unit is added to switch between the two driving modes based on the positioning error of the actuator.

Figure 8 shows the positioning performance in the dual driving mode. In the positioning, the reference consists of a 10 μm step signal and a 100nm stepwise input. The positioning error in the fine driving mode is smaller than 14 nm. This result demonstrates the high positioning performance and high adjustment performance near the reference position under the dual driving mode.

CONCLUSION

This paper introduced the dual driving mode control utilizing the wide driving mode for precision positioning in the wide working range and the fine driving mode for ultra-precision position adjustment near the reference position with the continuous holding force. This driving mode utilizes the relationship between the applied voltage signal and the friction force of the electrostatic actuator supported by only lubricating oil. The designed control system demonstrated the high ability to adjust position ultra-precisely with the continuous holding force. The positioning error is smaller than 14nm. The dual driving mode control is also useful for vertical stages.

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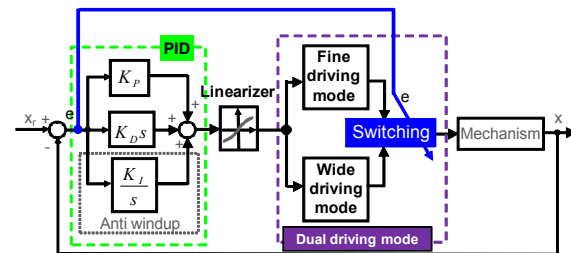


FIGURE 7. Block diagram of the control system for the PTP positioning.

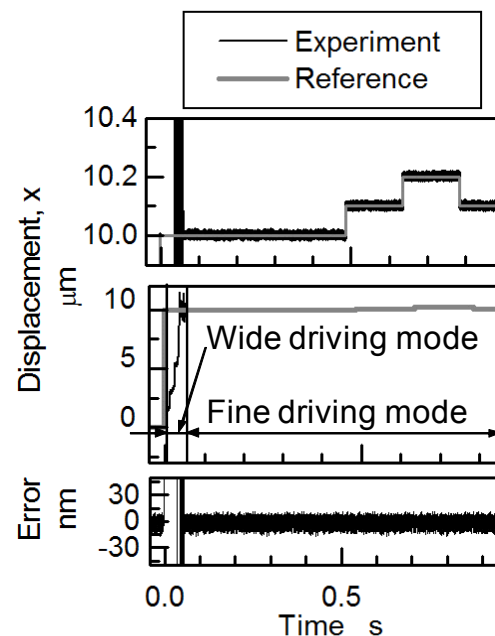


FIGURE 8. Positioning performance in the dual driving mode.

REDUCTION OF THE RESIDUAL VIBRATIONS OF ULTRA-PRECISION MANUFACTURING MACHINES VIA MODE COUPLING

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OVERVIEW

This paper presents a low-cost and energy-efficient method for reducing the undesirable residual vibrations that occur in ultra-precision manufacturing machines during the motion of their axes. An analytical model of a passively-isolated machine is used to demonstrate that coupling vibration modes by changing the isolator support points of the machine could provide optimal conditions for vibration reduction due to variations in the isolated system's natural frequencies and mode shapes. The proposed method is used in experiments to achieve a five-fold reduction of the dominant residual vibrations of an ultra-precision five-axis machine tool.

INTRODUCTION

Ultra-precision manufacturing (UPM) machines are designed to fabricate and measure complex parts having micrometer-level features and nanometer-level tolerances/surface finishes [1]. They enable the production of micro devices that are largely responsible for the advances in the electronics, biomedical, communications and other cutting-edge industries. Examples of UPM machines include ultra-precision machine tools, wafer steppers and micro CMMs, to name a few. Due to stringent accuracy requirements, UPM machines must be properly isolated from floor vibrations. Passive isolators provide an energy-neutral, reliable and cost-effective means for vibration isolation [2-4]. A major problem with the soft mounting provided by passive vibration isolators is that it induces low-frequency residual vibrations of the isolated machine base, which degrade the achievable accuracy/speed of UPM machines [2]. Consequently, to reduce residual vibrations, active vibration reduction systems are often employed [5] which leads to significant increases in the costs and energy consumption [2-4] of the machine.

The long-standing rule in isolation system design is to decouple all vibration modes by

aligning the isolator mounting locations with the center of gravity of the isolated machine [2,3]. Rivin [3], however, points out that the proper selection of the isolator mounting points as a means of reducing residual vibrations is an issue often ignored by designers. DeBra [2] makes a similar observation based on an ultra-precision diamond turning machine whose residual vibrations reduced when its center of gravity was raised relative to its isolator support point. However, neither DeBra [2] nor Rivin [3] provide an analysis-based explanation for the reduction in residual vibrations that occurs due to coupling the vibration modes of the isolated system by varying the isolator mounting points. This paper presents a preliminary analysis of mode coupling in passively-isolated systems and demonstrates its potential to achieve superior vibration reduction in UPM machines.

MODELING AND ANALYSIS

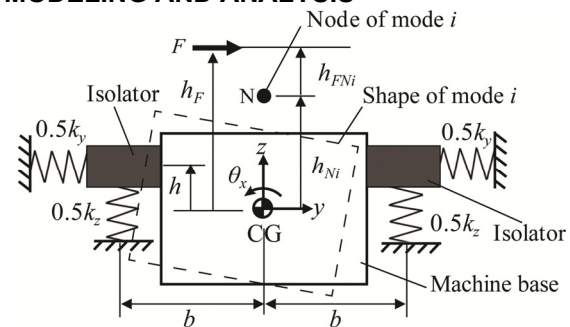


FIGURE 1. Simple Model of an Isolated Machine

Figure 1 shows a simple model of an isolated machine of mass m and centroidal moment of inertia I about the x -axis. k_y and k_z are the combined stiffnesses of the passive isolators in the y and z directions, respectively. The vibrations of the machine base are assumed to occur only in the y - z plane. Such planar analyses can be applied, for instance, when the machine's structure is symmetrical in the x -direction. Note that h is the vertical height of the isolator mounting point, measured from the

center of gravity (CG). Since the dynamics in the z-direction is not affected by h , it is not considered in the analysis. Our focus is on the two modes, described by the coordinates y and θ_x , which are coupled by non-zero values of h . Assuming, for theoretical convenience, that the system in Figure 1 is undamped, the equation describing the free vibrations of the system is

$$\mathbf{M}\ddot{\mathbf{u}} + \mathbf{K}\mathbf{u} = \mathbf{0} \quad (1)$$

$\mathbf{M}$, $\mathbf{K}$ and $\mathbf{u}$ are respectively the mass matrix, stiffness matrix and vector of displacements of the system. They are given by

$$\mathbf{M} = \begin{bmatrix} m & 0 \\ 0 & I \end{bmatrix}; \mathbf{K} = \begin{bmatrix} k_y & -k_y h \\ -k_y h & k_\theta + k_y h^2 \end{bmatrix}; \mathbf{u} = \begin{bmatrix} y \\ \theta_x \end{bmatrix} \quad (2)$$

where $k_\theta = b^2 k_z$.

The effect of the y-axis inertial reaction F on the acceleration (residual vibration) of the base in the θ_x direction is given by the frequency response function (FRF)

$$H(\omega) = \frac{-\omega^2 \theta_x}{F} = \sum_{i=1}^2 \frac{h_{Ni} - h_{Ni}}{I_{qi}} \frac{\omega^2}{-\omega^2 + 2j\omega\zeta_i\omega_{ni} + \omega_{ni}^2} \quad (3)$$

In Eq.(3), ω_{ni} , h_{Ni} and I_{qi} respectively denote the natural frequency, node height and modal inertia of each vibration mode i (where $i = 1, 2$), while ω represents the excitation frequencies; ζ_i is the modal damping added to account for the damping of the isolated system which was ignored in the modeling and j is the unit imaginary number. To understand how $H(\omega)$ is affected by mode coupling, the variation of ω_{ni} , h_{Ni} and I_{qi} as functions of h is studied based on their analytical expressions given by

$$\begin{aligned} \omega_{n1} &= \sqrt{\frac{1}{2mI}(A-B)}; & \omega_{n2} &= \sqrt{\frac{1}{2mI}(A+B)}; \\ h_{N1} &= h + \frac{k_\theta - \omega_{n1}^2 I}{k_y h}; & h_{N2} &= h + \frac{k_\theta - \omega_{n2}^2 I}{k_y h}; \\ I_{q1} &= mh_{N1}^2 + I; & I_{q2} &= mh_{N2}^2 + I; \\ A &= mk_\theta + mk_y h^2 + k_y I; & B &= \sqrt{A^2 - 4mIk_y k_\theta} \end{aligned} \quad (4)$$

The characteristic behavior of ω_{ni} , h_{Ni} and I_{qi} as functions of h is plotted in Figure 2 for two cases $k_y/m < k_\theta/I$ and $k_y/m > k_\theta/I$. As can be seen from Figure 2(a), as h increases positively or negatively, ω_{n1} reduces while ω_{n2} increases monotonically. At $h=0$, ω_{n1} and ω_{n2} come closest

to each other and then suddenly veer off with large curvatures without crossing. Crossing only occurs if $h=0$ and $k_y/m = k_\theta/I$.

The node height (h_{Ni}) is very important because it determines the moment arm of F (i.e., h_{FNi} in Figure 1). When $k_y/m < k_\theta/I$ (Figure 2(b)), mode 1 starts out as a purely translational mode ($h_{N1}=\infty$) at $h=0$. As h increases positively, the node height (h_{N1}) always remains positive but rapidly drops until the point $h^* = (k_\theta/k_y - I/m)^{1/2}$ after which h_{N1} starts to rise. For negative h , the same trend is seen but h_{N1} is always negative. Conversely, when $k_y/m > k_\theta/I$, mode 1 starts out as a purely rotational mode about the CG ($h_{N1}=0$). As h increases positively, h_{N1} rises above the CG monotonically while for negative h , it monotonically drops below the CG. Note that, irrespective of the system parameters, the node of mode 1 is always above the CG for positive h and below for negative h values. As shown in Figure 2(b), the trend for h_{N2} is opposite that of h_{N1} ; irrespective of the system parameters, the node of mode 2 is always below the CG for positive h and above for negative values of h .

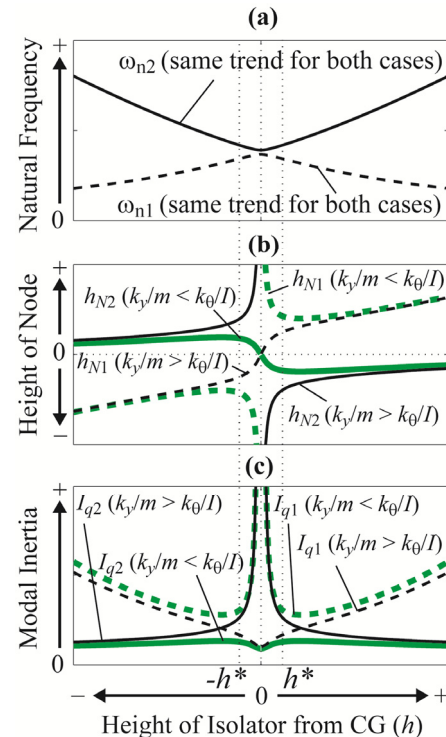


FIGURE 2. Variation of (a) Natural Frequencies (b) Height of Nodes and (c) Modal Inertias as Functions of Isolator Height h

Based on the relationships in Eq.(4), the modal inertias follow a similar trend as the node heights except that their behavior is symmetrical about $h=0$ (see Figure 2(c)). The modal inertia is lowest when the mode is purely rotational ($h_{Ni}=0$) and maximum (infinite) when the mode is purely translational. Generally speaking, a large modal inertia is preferable because it means that the mode has more resistance to excitation from external moments.

The just-described drastic behavior of ω_{ni} , h_{Ni} and I_{qi} as functions of h is related to closely-linked and well-studied phenomena known as eigenvalue curve veering and mode localization [6,7]. Curve veering and mode localization are described as catastrophic-type phenomena because small changes in the system parameters result in large variations in the eigenvalues and mode shapes in systems with (weakly) coupled vibration modes [6]. One very pertinent implication of curve veering and mode localization to machine isolation system design is their potential to create critical configurations in structures subject to broadband input, as first indicated by Igusa [7]. When critical configurations exist in a structure, small variations in parameters can lead to large variations in the response of the system. For example, resonance and anti-resonance are well-known critical configurations of harmonically loaded structures. Critical configurations can be useful in design if minor modifications will significantly reduce the response [7], as will be herein shown to be true for UPM machine isolation system design.

The variance of a given FRF $H(\omega)$ in response to a zero-mean, band-limited white noise input of unit magnitude is given by

$$\sigma^2 = \int_{\omega_{min}}^{\omega_{max}} |H(\omega)|^2 d\omega \quad (5)$$

It is a measure of the energy of the system's response to stationary random excitations [7] but also provides a qualitative indicator of the system's transient response magnitude. In the simulations and experiments that follow, we will demonstrate that a judicious selection of h can lead to significant reductions in the variance of the residual vibration FRF given in Eq.(3).

SIMULATIONS AND EXPERIMENTS

Simulations and experiments are carried out on Mori Seiki's NN1000 ultra-precision 5-axis machine tool prototype [8] whose schematic is shown in Figure 3. Stemming from a granite base supported on four pneumatic vibration isolators, the machine has its x and C axes on the workpiece side and its y , z and A axes on the tool side. The key parameters of the machine, used for simulations, are provided in Table 1.

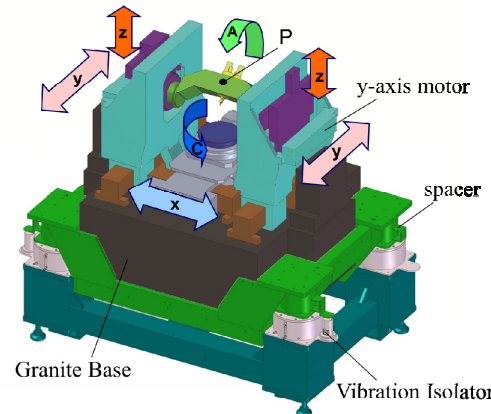


FIGURE 3. Schematic of Mori Seiki's NN1000 5-Axis Ultra-Precision Machine Tool Used for Simulations and Experiments

TABLE 1. Simulation Parameters

Parameter	Value
m [kg]	1,182
I [kgm ²]	96
k_y [N/mm]	880
k_θ [Nm/rad]	104,430
h_F [mm]	435

Figure 4(a) shows a plot of the normalized variance of H calculated from Eq.(5) with integration limits $\omega_{min} = 0$ and $\omega_{max} = 62.8$ rad/s (10 Hz), using the parameters from Table 1 and a fixed empirical damping ratio $\zeta_i = 15\%$ for both modes. Within the range of interest (i.e. $h = -150$ to $+150$ mm), the variance is lowest when the CG is highest with respect to the isolator support point (i.e. $h = -150$ mm). Figure 4(b) shows the simulated FRF magnitude while Figure 4(c) shows the simulated unit step response of H for $h = -116$, 0 and 116 mm. The height of 116 mm is selected based on the maximum height restrictions of the machine. As predicted by the variance curve, the highest allowable CG (i.e., $h = -116$ mm) has the smallest vibration peaks in frequency domain and the least amount of vibration in time domain.

In experiments two isolator heights are considered, i.e., $h = 0$ and $h = -116$ mm. The CG could not be lowered below $h = 0$ because of design restrictions. Figure 5 shows the measured magnitude of $H(\omega)$ as well as the y-acceleration of the machine, measured from the point marked P on Figure 3, during the motion of the y-axis. Raising the CG reduces the measured residual vibrations significantly leading to a five-fold reduction in the RMS position error measured from the y-axis encoder during the motion of the y-axis, as shown in Figure 6. Simulation results reported in [9] show that mode coupling can also simultaneously improve vibration isolation quality (i.e., reduce transmissibility).

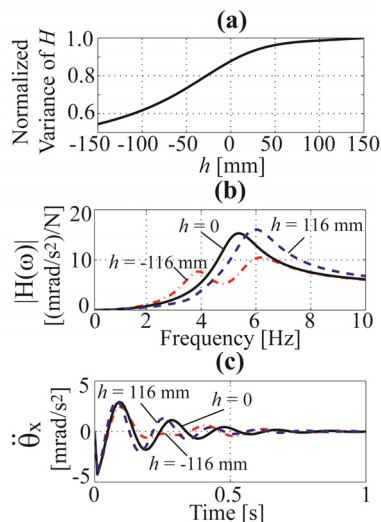


FIGURE 4. Simulated (a) Normalized Variance (b) FRF Magnitude (c) Unit Step Response of NN1000

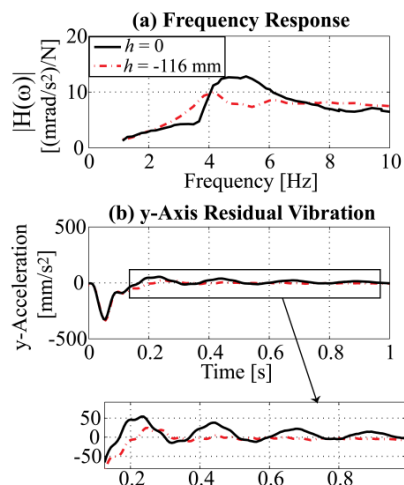


FIGURE 5. Measured (a) FRF Magnitude (b) y-Axis Residual Vibrations of NN1000 DCG

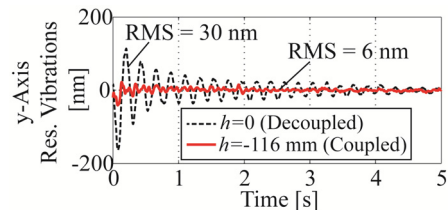


FIGURE 6. Measured y-Axis Position Error (from Position Encoder)

CONCLUSION

This paper has shown analytically that coupling the vibration modes of passively-isolated systems by adjusting the location of the isolator mounts could provide conditions that lead to reduced residual vibrations. This method of residual vibration reduction is very attractive because it is simple, cost-effective and energy-efficient compared to active vibration reduction techniques. A five-fold reduction of the dominant residual vibrations of an ultra-precision machine tool is demonstrated in experiments.

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CONTROL SCHEMES FOR MICROFLUIDIC VIRAL DNA/RNA AMPLIFICATION

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INTRODUCTION

The polymerase chain reaction (PCR) has become the standard technique for genetic detection and identification with applications ranging from clinical diagnostics to forensics [1-4]. PCR allows highly specific and sensitive detection of target sequences by exponentially amplifying a specific region of DNA or RNA from as little as a single copy through thermocycling a biochemical cocktail. While traditional PCR, based on either conductive or convective heating for the requisite thermocycling [5,6] has been an adequate tool thus far, we aim to decrease the cost and time associated with PCR by direct radiative heating of an aqueous sample in a polymer microfluidic chip. Our thermocycling technology relies on a 700 mW 1450 nm infrared laser diode for direct heating of the sample. A reaction volume of 1 μL , smaller than conventional reactions by 5-50x, decreases the consumption of costly reagents and allows for rapid cycling with heating rates 2-10x faster than conventional instruments. The reactions take place inside disposable polymer microchips that are directly micro-milled for accurate dimensions and repeatability [7]. Our lab has developed infrared radiation-based PCR thermocyclers with both open and closed-loop control schemes, all operated through user-friendly Labview interfaces. The different control methods each offer unique advantages suited to various applications, including rapid, quantitative PCR as well as high-throughput screening of multiple distinct genetic targets.

OPEN-LOOP CONTROL

A major challenge for implementing a low-cost polymer microchip for microfluidic PCR is accurate temperature measurement. This often requires sensors in direct contact with the aqueous PCR sample, since the relatively low thermal conductivity of the microchip material prevents the device from reaching thermal equilibrium with its environment. This is especially true for a system using infrared

heating, since heat transfer is localized to only the reaction volume. Another complication of direct contact between thermocouples and the aqueous PCR sample is interference with the reaction chemistry. One way to avoid these challenges is to utilize open-loop control where there is no active temperature measurement requirement. A method for open-loop control offers the benefits of simple design, faster setup time, and elimination of contact temperature sensors while retaining the advantages of rapid radiative heating of a small volume. In order to obtain these benefits and implement open-loop control, excellent repeatability and accurate calibration prior to running PCR are crucial [8].

To ensure excellent repeatability, an environmental control fixture (shown in Fig. 1) was fabricated for mounting the microchip to achieve precise positioning and environmental stability. For positioning, the fixture aligned the microchip to the infrared laser using 1/16 inch dowel pins, which were press fit into a block that interfaces with the laser. Corresponding alignment holes were drilled and reamed into the microchips during manufacturing. For environmental stability, the fixture was machined from tellurium copper and features internal water cooling from a circulating bath to keep the area directly surrounding the chip at a known constant temperature, eliminating the effects of ambient temperature disturbances. Additionally, the fixture allows the delivery of 40 psig nitrogen to the fill ports of the microchip, enabling the chamber to remain pressurized during thermocycling to prevent the formation and expansion of disruptive air bubbles during the PCR reaction. To observe positioning repeatability, the system was mounted on an inverted microscope (Nikon, TE2000-E). An empty microchip was mounted in the fixture and imaged with the microscope, recording the position of the chamber. This process was then repeated for 10 trials. To test thermal stability, a micro-thermocouple (Physitemp, T240C) was

bonded into the microchip, which was filled with water, mounted into the system, and temperature was monitored for 1 min. These mechanical and thermal repeatability tests yielded an average variation in chip placement of 5 μm , or 1% of the reaction chamber width, and the standard deviation in reaction chamber temperature was found to be 0.076°C when maintaining a constant temperature.

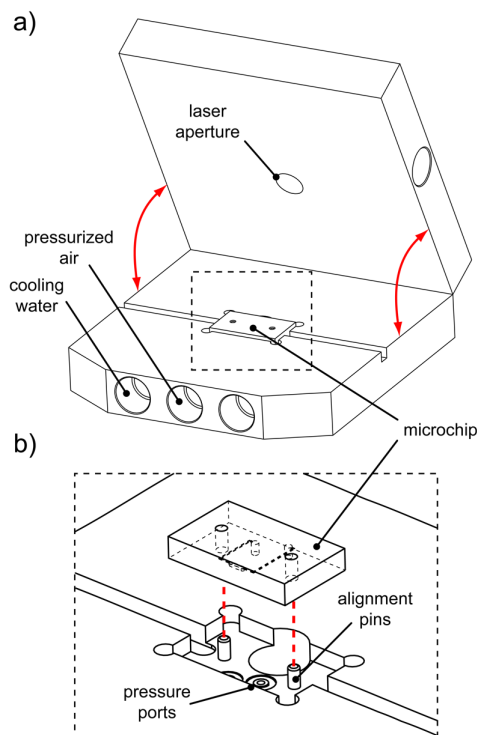


FIGURE 1. The environmental control fixture provides a mechanically and thermally repeatable environment for the microchip (inset) while allowing microscope and laser access to the PCR sample contained in a chamber in the center of the microchip.

In order to implement open-loop control, a calibration relating laser driving voltage to aqueous solution temperature is necessary. Since the thermocouple used for calibration is heated directly by the laser, we compensated for this bias error as follows. First, it was known that the threshold voltage for the laser to turn on was 0.25 V. At this driving voltage the bias is zero and the thermocouple reads a temperature of 33°C, identical to the chamber temperature. Next, the driving voltage was increased as the reaction chamber was observed with a microscope. Boiling, as indicated by the rapid

formation and expansion of bubbles in the chamber, occurred at 1.1 V and therefore corresponds to 100°C at atmospheric pressure. With the thermocouple centered and positioned at the bottom of the chamber, the measure bias increases from 0°C at 0.25 V to around 8°C at 1.1 V. The variables affecting this bias include thermocouple placement repeatability, bubble formation, and direction the thermocouple tip is bent. A calibration microchip was manufactured with the thermocouple bonded to the bottom layer (farthest from the laser). This prevents many of these variations between runs because the thermocouple tip is in the same position for every trial. Also, since our thermal modeling predicted that 90% of the radiation is absorbed in the first 0.5 mm of the aqueous solution, confounding effects of radiative heating of the thermocouple directly were minimized with the thermocouple located at the bottom of the chamber. A polynomial was fit to the bias corrected temperature measurements, resulting in a final calibration curve.

An optimal laser power profile for thermocycling was then determined using the calibration data along with calculations using an equivalent thermal circuit to model the heat conducted from the chamber. A lumped capacitance model is also included to model the transient temperature response due to the dynamic heating and cooling. An optimizer then modifies the laser power input to the circuit model to attain the desired three temperatures necessary for PCR: denaturing at 94°C, annealing at 68°C, and extension at 72°C. The performance of the calibrated system is shown in Fig. 2.

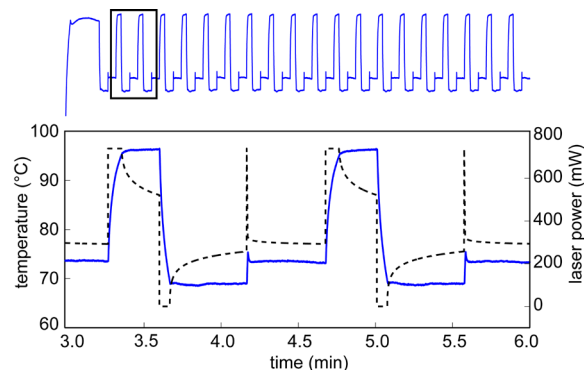


FIGURE 2. Using a calibration of laser power and reaction temperature combined with heat transfer modeling, an optimized laser power profile (dashed line) was generated for open-loop thermocycling.

The exponentially increasing and decreasing laser power profiles produced stable chamber temperatures and minimized the temperature undershoot and overshoot of the system. During one thermocycling run with 20 cycles, we measured the average standard deviation of the 20 respective denaturing, annealing, and extension steps to $\sigma_d = 0.16^\circ\text{C}$, $\sigma_a = 0.18^\circ\text{C}$, and $\sigma_e = 0.15^\circ\text{C}$, respectively. In addition, the average heating and cooling rates of the system were $3.3^\circ\text{C}/\text{sec}$ and $3.86^\circ\text{C}/\text{sec}$, respectively. To determine the run-to-run variability we measured the average denaturing, annealing, and extension temperatures over all 20 cycles for a run and compared these three numbers to another run. The average absolute difference between these temperatures was $\mu = 0.12^\circ\text{C}$.

CLOSED-LOOP CONTROL

While open-loop control allows for a simplified instrument design and faster setup time, temperature feedback, or closed-loop control, can be useful for more complex aims such as independently controlling the temperature of multiple reaction chambers on the same microfluidic chip. This is valuable for amplifying multiple different DNA sequences in parallel, requiring different annealing temperatures, simultaneously on the same chip from a single or multiple laser sources. Furthermore, closed-loop control allows for some variability in manufacturing and setup and accommodates disturbances from the environment.

In conventional convective or conductive thermocyclers, the common method of temperature measurement is to embed a thermocouple in the rough proximity of the sample container and allow the environment to thermally equilibrate. However, direct, rapid heating such as with laser radiation makes this method more challenging in that a very accurate model for the relation between solution temperature and measured thermocouple temperature is required.

To correlate the chip temperature with the solution temperature in our work, a calibration chip is used that has two thermocouples: one inside the reaction chamber and one embedded in the chip. By driving the laser at different powers and waiting for the calibration chip to reach steady state, a relationship is determined between the solution temperature and chip temperature. This relationship is then used in a real-time control algorithm for closed-loop

control of the laser. A Labview program implementing this algorithm computes the difference between the measured and desired temperatures and uses a proportional-derivative (PD) controller to determine the laser power.

In the case of a single laser source, the beam is collimated and split with a lens array to focus radiation on adjacent chambers. A solenoid shutter is positioned below one of the chambers to attenuate the radiation hitting that chamber with actuation using pulse width modulation (PWM) control. By varying the duty cycle at which the solenoid is modulated, the temperature of that chamber can be effectively decreased up to about 10°C for chambers spaced 1 mm apart. For multiple laser sources, a single laser driver operating them in tandem can be used with multiple independently controlled solenoid shutters. This case, dual lasers and shutters driven by a common laser driver, is depicted in Fig. 3.

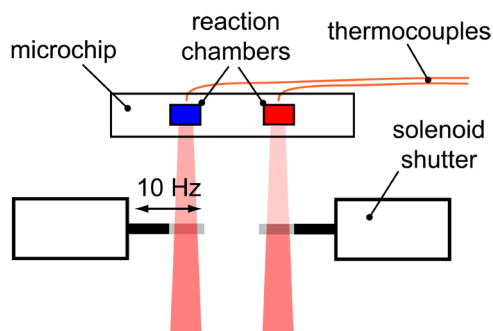


FIGURE 3. Infrared laser sources and solenoid shutters can be used with closed-loop control to perform PCR on multiple distinct targets simultaneously by monitoring temperature and independently modulating the laser beams.

The resulting capability of independent temperature control for two different reactions is shown in Fig. 4. By consolidating multiple thermocycling runs into a single step while using small sample volumes and harnessing the rapid direct heating by laser radiation, there is potential for massive cost and time savings. In addition, this configuration is scalable for increasing throughput by adding additional laser sources or dividing a single, more powerful laser and coupling each path with a shutter.

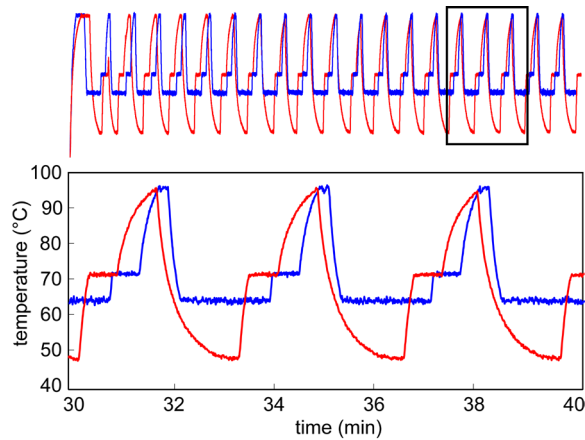


FIGURE 4. Closed-loop control and optical modulation enable independent temperature profiles for amplifying multiple genetic targets in a single instrument run.

CONCLUSION

The control schemes presented here coupled with the high-throughput, small volume, rapid cycling nature of infrared heating enable quick results in time sensitive applications. Open loop control is advantageous in that it offers simple and fast setup and execution, but it is most suitable for applications requiring only single reactions that are well calibrated. Open loop control is more complex and costly to implement and operate, but can allow multiple simultaneous chambers and accommodate environmental disturbances. Both open and closed-loop control have unique advantages, and the combination of a compact, portable system with rapid cycling times and disposable chips makes this technology ideal for point-of-care scenarios.

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A NEW HYBRID CONTROL STRATEGY INCORPORATING FAST-TOOL SERVOS

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INTRODUCTION

Continuing desire for higher productivity in precision applications has led to the increasing use of various types of "fast-tool servos" in these applications. These mechanisms, including galvanometers, piezoelectric actuators, and voice-coil motors, have very high bandwidth, but usually very limited range of motion. This means that they must be coupled somehow with mechanisms that have a greater range of motion in order to cover the full range of motion required of the machine in each dimension. The longer-range mechanisms, often driven by standard electromagnetic servomotors, typically have significantly lower bandwidth, but can cover the full required range of travel.

Deciding which components of motion should be allocated to the fast-tool actuators and which to the standard actuators has not been an easy task. A common strategy has been to use the standard axes just to move the fast-tool axes into range, then hold them still while the fast-tool axes do the actual operation in that zone. This approach significantly limits throughput and flexibility.

The problem is compounded by the common desire to use automatically generated tool-tip paths from standard CAD/CAM software that has no knowledge of these hybrid mechanisms, or algorithms to deal with them. This often is not possible, leading designers of machines of this type to attempt to either write their own specialized CAD/CAM software, or to create post-processing software to take standard CAD/CAM output files and create hybrid files that allocate different portions of the motion to the fast and slow axes. In either case, this software must have very precise knowledge of the algorithms in the controller(s) in order to create a proper net tool-tip path at all times.

A further complication is encountered if the machine has a complex relationship between the tool-tip coordinates and the underlying actuator positions. This relationship, known as the kinematic transformation, is more complex than a simple scaling and offset on an increasing number of machines these days. This adds more difficulty to the algorithms that allocate different components of motion to fast and slow actuators.

Sometimes, the fast and slow axes are controlled by different devices, with different update rates. This can be very difficult to coordinate properly, both in terms of the times for programmed moves (which are usually specified in terms of speed, not time) and the "heartbeat" of the controller's own time base.

A new process overcomes these problems, permitting standard part programs from commercially available CAD/CAM software to be used, and processed in real time in the controller to allocate the proper components of motion to the standard and fast-tool actuators, keeping the net tool-tip path to what is programmed at all times.

STRATEGY CONCEPT

This process combines two key ideas. First is a "spectral decomposition" of the trajectory into low-frequency and high-frequency components for use by the standard and fast-tool axes, respectively. Second is strategic conversion between common tool-tip coordinates and the actuator coordinates for the two mechanisms so the two components can be properly combined in a synchronized fashion.

Figure 1 shows the strategy in block diagram form. Each step will be explained in detail.

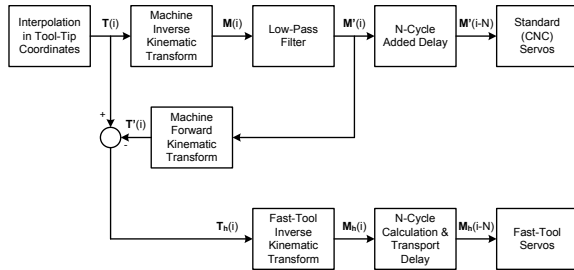


FIGURE 1. Block diagram of hybrid machining strategy

STRATEGY IMPLEMENTATION

The process is implemented in a multi-step sequence starting from a standard description of the tool-tip path. This description is typically an RS-274 “G-code” part program, as generated in standard CAD/CAM software.

Preparatory Step: Create a Part Program in Tool-Tip Coordinates

The first step is to create a part program that describes the tool-tip path in the coordinates of the tip relative to the part, regardless of the underlying geometry of the machine. The feedrates (speeds) of the tool-tip should be those attainable by the hybrid machine, and if the part program description determines any accelerations, these should be what the hybrid machine (dominated by the fast-tool) can accomplish. Fundamentally, the program is created for an “ideal” machine that could execute the paths directly at full speed and bandwidth.

Many standard CAD/CAM software packages can create this type of part program directly, and the ability to use these commercially available packages is one of the key advantages of this procedure.

Step 1: Perform Initial Processing of Part Program as if for “Ideal” Machine

The controller will perform the initial processing of the part program as if the perfect machine that could execute this tool-tip path with one actuator per coordinate, directly mapped, existed. This can be standard CNC-style processing, creating equations of commanded motion from the move descriptions, then solving these equations every servo update cycle to calculate a set of net tool-tip coordinates for that cycle (which we will call the vector $T(i)$, with “i” being the servo cycle index number).

Step 2: Convert Tool-Tip Coordinates to Standard Actuator Coordinates

The next step is to convert the tool-tip coordinate vector $T(i)$ each cycle to the “motor” coordinate vector $M(i)$ that contains the positions of the actuators on the standard (“slow”) mechanism that would be required to obtain these tool-tip coordinates, assuming that the fast-tool mechanism is held in its reference position. This conversion is known as the inverse-kinematic transformation.

In some mechanisms, such as a Cartesian table, this transformation could be a simple scaling and offset. In other mechanisms, such as 4-axis and 5-axis CNCs, Stewart platforms (“hexapods”), and robotic arms, this transformation could be a complex geometric conversion. In most cases, there are as many actuators as controlled tool-tip coordinates, so the vectors $T(i)$ and $M(i)$ are of the same length.

To this point, the process is operating as if the standard mechanism had the responsiveness to execute the part program by itself, with the fast actuators holding still in their reference position.

Step 3: Filter the Actuator Coordinate Trajectory

Once this ideal (and unexecutable) trajectory has been calculated for the standard mechanism, this trajectory is passed through some form of digital low-pass filter, creating a filtered command trajectory that is within the capabilities of the standard mechanism. The cutoff frequency of the filter should be within the bandwidth of the mechanism. Technically, each actuator in the multi-axis mechanism will have its own independent filter running in parallel with the filters for the other axes. The filters for each axis do not have to have the same characteristics, such as cutoff frequency, but often will.

In practice, these filters will likely be simple IIR (infinite impulse response) filters for computational efficiency, due to the high update frequencies at which they will be executing. Each cycle’s output would be a weighted average of the cycle’s unfiltered input from the previous step and one or more of the previous cycle’s filter outputs. However, other types of filters are possible here as well. The resulting output vector of filtered motor coordinates will be called $M'(i)$.

Step 4: Create an N-Cycle Deep Buffer

Because there can be delays in transporting and processing the data for the control of the high-bandwidth axes (especially if, as is often the case, these are controlled by a separate controller), the algorithm needs to be able to create a comparable delay in the trajectory to the standard axes. To do this, a simple buffer data structure can be created in software in which each servo cycle the new vector $\mathbf{M}'(i)$ is input to the buffer, and an N-cycle-old vector $\mathbf{M}'(i-N)$ is output from the buffer.

Step 5: Send Processed Coordinate Set to Standard Axes

Steps 1 through 4 have computed the component of the commanded trajectory from the ideal part program that the standard axes can execute. These coordinates are fed to the servo loops of the standard axes as set points.

In many, if not most, of these applications, the loop closure rate of these standard axes does not need to be as high as that for the high-bandwidth axes. Therefore, it will be desirable to be able to skip update cycles between consecutive servo loop closures on the standard axes, both to conserve computational resources and to let the loop closure rate be more appropriate for the dynamics of the standard axes. When cycles are skipped between successive closures, it is still important to compute new setpoints every cycle, as these will be needed in the creation of the trajectory for the high-bandwidth actuators.

Step 6: Convert Filtered Actuator Coordinates Back to Tool-Tip Coordinates

The filtered trajectory computed for the standard actuators is in the coordinates of the actuators themselves, not those of the tool tip. To determine the difference between the net tool-tip path and the filtered tool-tip path, which is required to compute the component for the high-bandwidth actuators, this filtered actuator setpoint vector $\mathbf{M}'(i)$ must be converted back to a filtered tool-tip setpoint vector $\mathbf{T}'(i)$. This conversion is known as the forward-kinematic transformation. In this process, it will be performed every servo update cycle "i". Note that this conversion operates on the actuator setpoint vector before the delay (i.e. the output from Step 3, not from Step 4).

As with the inverse kinematic transformation, this could be as simple as a scaling or offset, or in the case of non-Cartesian mechanisms, a much more complex geometric conversion. In most cases, there are as many actuators as controlled tool-tip coordinates, so the vectors $\mathbf{M}'(i)$ and $\mathbf{T}'(i)$ are of the same length.

Step 7: Compute Fast-Tool Component of Tool-Tip Trajectory

From Step 1, we have the ideal tool-tip coordinates from the standard interpolation algorithms. From Step 6, we have the filtered tool-tip coordinates attainable by the standard actuators. The high-bandwidth actuators must make up the difference. The setpoint vector $\mathbf{T}_h(i)$ for the high-bandwidth actuators can simply be calculated by subtracting the filtered tool-tip setpoint vector $\mathbf{T}'(i)$ from the ideal vector $\mathbf{T}(i)$: $\mathbf{T}_h(i) = \mathbf{T}(i) - \mathbf{T}'(i)$. This calculation must be performed every servo cycle "i".

It is possible that the high-bandwidth mechanism will not have as many degrees of freedom as the tool-tip and standard coordinate sets. In these cases, only part of the high-bandwidth tool-tip vector $\mathbf{T}_h(i)$ will be used.

Step 8: Convert High-Bandwidth Tool-Tip Coordinates to Actuator Coordinates, Transmit to Controller

Once the high-bandwidth tool-tip setpoint vector $\mathbf{T}_h(i)$ has been computed, one or two more steps must be performed, in a manner that is dependent on the particular configuration. The vector $\mathbf{T}_h(i)$ must be converted to the coordinates of the high-bandwidth actuators, using the inverse-kinematic transformation for the high-bandwidth mechanism (which can be completely independent of the kinematics for the standard mechanism), yielding an actuator setpoint vector $\mathbf{M}_h(i)$. Depending on the mechanism, this can be a simple scaling and offset for each axis, or a more complicated transformation. The transformation can be algorithmic or table based.

In some cases, the high-bandwidth actuators are controlled by a separate controller from the standard actuator. In these cases, the setpoints for the high-bandwidth actuators must be transmitted to this separate controller, usually resulting in a delay that requires comparable buffering in Step 4. The setpoints may be in the tool-tip coordinates, in which case any inverse-

kinematic transformation must be performed in the high-bandwidth controller, or in the actual actuator coordinates, in which case any inverse-kinematic transformation would have to have been performed first in the standard controller.

SAMPLE IMPLEMENTATIONS

Single-Axis, Dual Parallel Actuators

In this simple but commonly used case of a linear fast-tool actuator riding on top of a standard actuator, the geometric issues are trivial, but the separation of the desired tool-tip trajectory into high-frequency and low-frequency components is still used.

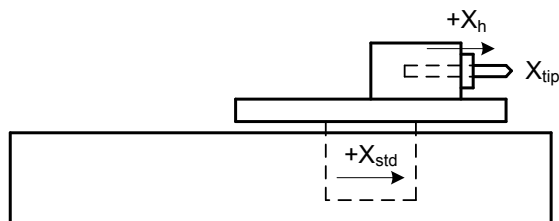


FIGURE 2. Dual parallel actuators

Cartesian Table with Galvanometer Laser System

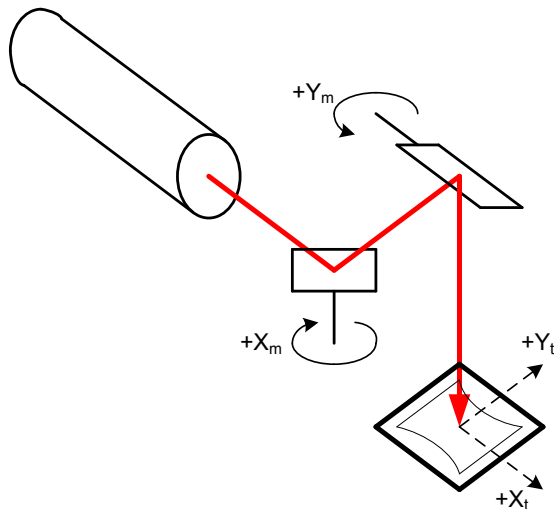


FIGURE 3. Galvanometer-controlled laser beam geometry

Another common configuration is a Cartesian table with standard actuators, and a laser beam deflected by two galvanometer-controlled mirrors acting on an object on the table. The geometry of the laser path to the table

introduces some kinematic transformation issues.

5-Axis CNC with Galvanometer Laser System

A more complex system, and the one that motivated this technique, involves a 5-axis CNC holding the part to be machined, and a laser-beam deflected by two galvanometer-controlled mirrors acting on the held part.

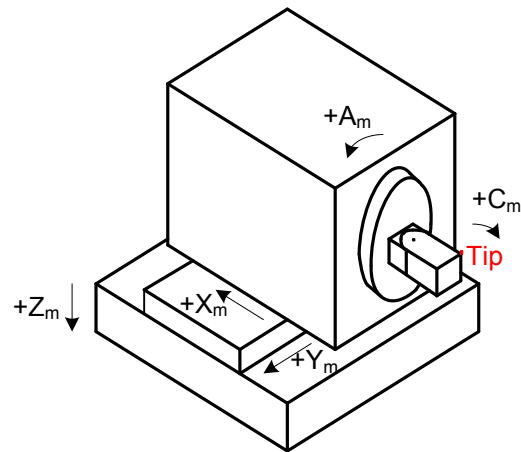


FIGURE 4. 5-Axis CNC Geometry

RESULTS

The result of this multi-step process is fully coordinated motion between the standard mechanism and the fast-tool mechanism, with the standard mechanism only doing motions within its bandwidth of operation, and the fast-tool mechanism only doing motions within its limited range of motion. All of this can be done automatically, utilizing standard part programs that do not need to consider the division of motion, and machine-specific algorithms that are independent of any particular part program.

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WIDE BANDWIDTH CONTROL OF BALLSCREW DRIVE USING CENTER OF MASS LOCATION FEEDBACK

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INTRODUCTION

Ballscrew mechanisms are commonly used in many precision positioning devices to convert rotary motion to translational motion. The manufacturing tolerances of a ballscrew play a significant role in the motion accuracy it can provide. However, in highly dynamic applications, the motion delivery of a ballscrew is not only influenced by the accuracy grade it has, but more importantly by the way it deforms and vibrates under the inertial loads and other forces. Although the challenge posed by the structural flexibilities can be dealt with by enhancing the system damping through mechanical re-design and modifications, a more robust and cost-effective approach would be to model these effects mathematically and try to compensate them by applying well-designed controls.

In general, the existing flexibilities not only affect the positioning accuracy, but also the servo bandwidth achievable when tuning the controller parameters. To increase the bandwidth to the maximum possible, we try to actively dampen out the first mode of vibration. Considering the fact that the eigen-frequencies of a ballscrew-driven machine are typically well separated, this can lead to a significant improvement in terms of control performance and motion accuracy. As a result, the two-mass model which can sufficiently represent the first mode of vibration has become popular and widely adopted in the literature addressing the control of flexible manipulators and feed drives.

In this paper, we also adopt a two-mass model and develop a control strategy which is based on decoupling the rigid body mode of motion from the elastic mode. Since the differential equation governing the rigid body mode is similar to one of a double integrator, the location of the center of mass (COM) which is representative of the rigid body motion can be controlled easily. The vibrations associated with the elastic mode are also suppressed by

injecting active damping proportional to the velocity difference between the two masses into the system. It is shown that the proposed controller materializes the goals of good command following and disturbance rejection on the load side by enforcing the center of mass to follow a desired trajectory compatible with the reference trajectory, and dampening out the elastic mode of motion simultaneously.

LITERATURE REVIEW

Alleviating the detrimental effects of resonance in position and velocity control of flexible manipulators and machines has been the subject of extensive research since 1990's. The proposed solutions for this problem range from simple methods well received by the industry such as the acceleration feedback method [1] to more complicated and sophisticated ones including among others: model predictive control [2], adaptive sliding mode control [3], and pole-placement control [4]. In [5], the authors applied the concept of sensor averaging to decouple the translational rigid body mode of motion from the torsional elastic mode. They then used separate control strategies for positioning the table, and for minimizing the yaw vibrations in the T-type gantry. In this paper, similar idea has been applied to a ballscrew drive to minimize the vibrations in the axial direction.

DESIGN METHODOLOGY

The main controller is designed assuming a two mass model for the ball screw drive as shown in Fig 1. The location of center of mass in this model is determined by:

$$\bar{x} = G_{\bar{x}}(x_1, x_2) = \frac{m_1}{m_1 + m_2} x_1 + \frac{m_2}{m_1 + m_2} x_2 \quad (1)$$

Fig. 2 compares the measured FRF of a single axis ball screw drive with the analytical FRF fitted to it based on the two-mass model assumption. As can be seen this system has a

resonance at around 135 Hz which is accurately represented by the fitted two-mass model. In addition, it reveals other resonances at higher frequencies which cannot be captured by the two-mass model.

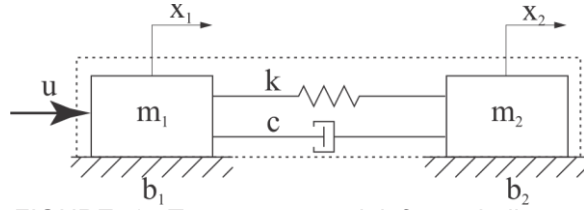


FIGURE 1. Two mass model for a ballscrew drive.

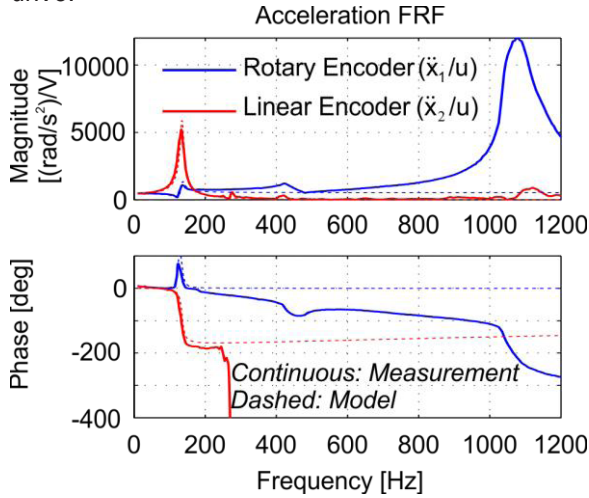


FIGURE 2. Measured FRF versus two-mass model FRF

If the viscous friction forces external to the system are neglected (i.e. assuming $b_1 = b_2 =$

0), the acceleration of the center of mass turns out to be proportional to the input force (u):

$$u = (m_1 + m_2)\ddot{\bar{x}} \quad (2)$$

This is due to the fact that the spring and damper forces are internal to the system and have no influence on the location of the COM. We later on consider the effect of these forces by treating them as external disturbances to the system.

Eq. 2 describes the dynamics of a double integrator. The idea here is to use a control law such as a simple PID to enforce the COM follow a trajectory ($\bar{x}_r$) consistent with the second mass following the reference trajectory (x_r):

$$\bar{x}_r(s) = \left[\frac{m_2}{m_1 + m_2} + \frac{m_1}{m_1 + m_2} \frac{m_2 s^2 + (b_2 + c)s + k}{cs + k} \right] x_r \quad (3)$$

In order to minimize the internal vibrations of the two-mass system, a control force proportional to the velocity difference between the two masses was also applied to the system:

$$u = C \frac{d(x_2 - x_1)}{dt} = C \frac{d(\Delta x)}{dt} \quad (3)$$

$$\Rightarrow U = Cs\Delta X$$

It can be shown that applying the active force described in Eq 3 increases the damping ratio without altering the natural frequency of the oscillations.

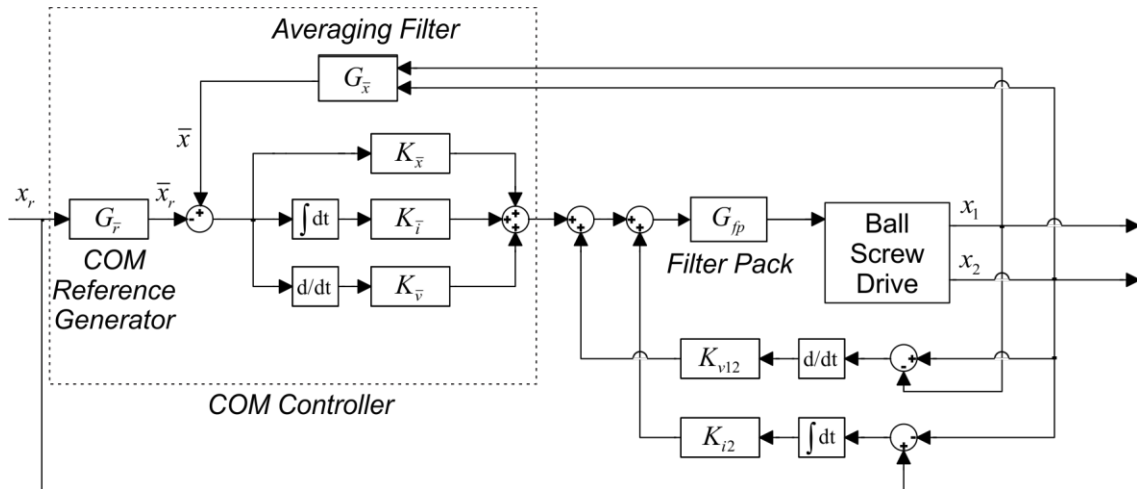


FIGURE 3. Overall control scheme.

To better control the load side of the two-mass system and push the steady-state tracking error to zero, an integral controller was also implemented in parallel with the COM controller. Fig 3 shows the overall control scheme that has been implemented in this work. To mitigate the effect of higher-frequency resonances on the stability of the designed controller, a filter pack comprising of a low pass filter and two notch filters was added to the feedback loop. The averaging filter in Fig 3 was used to calculate the feedback signal according to Eq 1 and send it to the COM controller.

Loop shaping techniques [6] were then used for tuning the parameters of the controller. The objective was to increase the cross over frequency of the servo loop to the maximum possible while maintaining at least $PM=35^\circ$ of phase margin. The control laws were implemented on a dSpace platform using sampling frequency of 20 kHz. The magnitude and Nyquist plot of the servo loop are shown in Fig. 4 and Fig 5 respectively. As can be seen in Fig. 4, the maximum cross over frequency that could be achieved was 115 Hz. It is also evident from Fig. 5 that the discretization of the control law slightly reduces the phase margin that could be otherwise achieved with a continuous implementation.

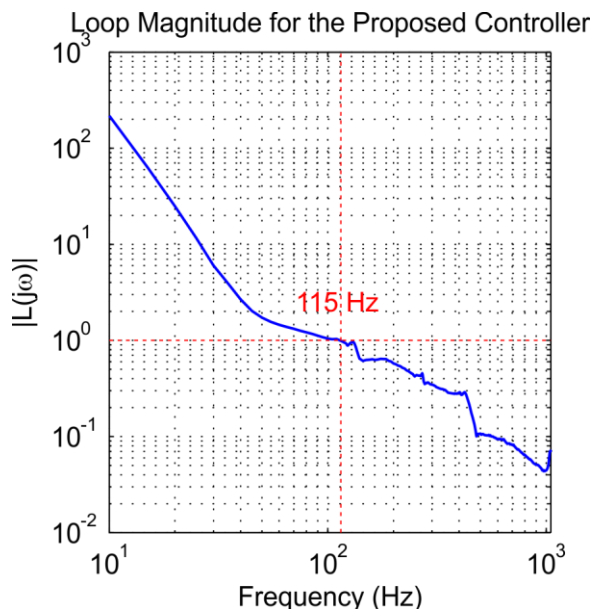


FIGURE 4. Loop magnitude plot for COM.

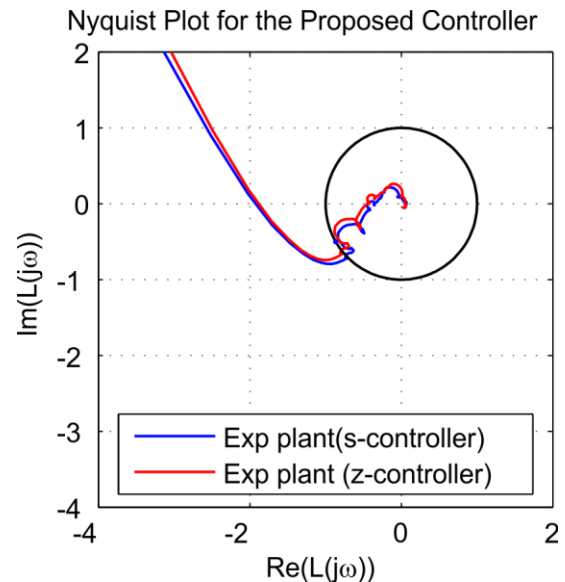


FIGURE 5. Nyquist plot for proposed controller.

Since the internal stiffness of the two-mass model does not influence the rigid body motion response of the system, robust stability against variations in drive stiffness is expected for a COM controller. To investigate the effect of parameter uncertainties on the stability of the proposed controller, a set of time-domain simulations were conducted in which the drive stiffness (k) and load mass (m_2) were perturbed according to a test matrix, and the stability was then evaluated based on the simulated responses.

Fig 6 demonstrates in blue the range of parameter variations that leads to a stable system with a COM controller. The COM controller used in these simulations did not include a load-side integrator ($K_{i2}=0$). As can be seen this controller tolerates a wide range of parameter variations both in drive stiffness as well as the load mass. A similar study was also carried out for a Pole-Placement Controller (PPC) designed previously for the same setup. The range of stable variations in parameters for PPC is identified by the red color in Fig 6. As can be seen, although the PPC provides acceptable robustness against parameter variations, the advantage of COM over PPC is obvious in this regard.

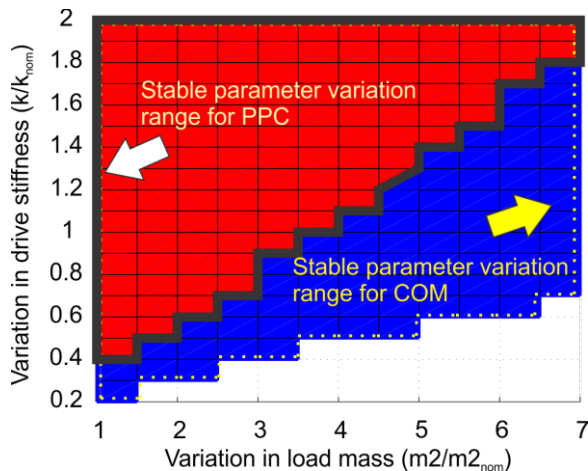


FIGURE 6. Simulated stable parameter variation domains for PPC and COM.

EXPERIMENTAL RESULTS

The capability of the proposed controller in rejecting disturbances was tested by conducting an impact hammer test on the table while it was under closed loop position control. Fig 7 shows the measured values of the disturbance FRF before and after the PPC is replaced by the COM controller. The results indicate that the disturbance rejection performance of the COM controller is not only as good as of the PPC controller in a wide range of frequency, but it even outperforms the PPC around the frequency that the magnitude of the disturbance FRF is maximum.

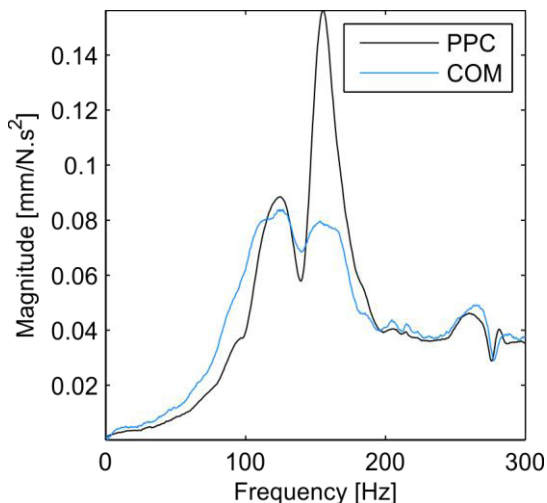


FIGURE 7. Disturbance FRF measured through impact hammer testing for COM and PPC controllers.

High-speed tracking tests were also conducted to verify the command following performance of the COM controller. The tracking error results for a continuous jerk 340 mm reference trajectory with 1 m/s velocity, 10 m/s² acceleration and 200 mm/s² jerk are shown in Fig 8.

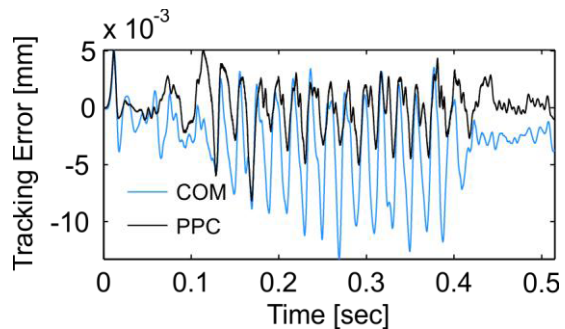


FIGURE 8. Comparison of trajectory following between the PPC and COM

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Recent Advancements in Motion Control with Long-Term Nanoscale Stability, Including a Novel, Comparative Validation Technique

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ABSTRACT

Advancing needs for positional resolution in semiconductor microlithography, data storage and related industries have been accompanied by commensurate needs for positional stability. For example, a great deal of technological development and facilities investment has been devoted to devising and deploying vibration isolation technologies which shield sensitive equipment from environmental perturbations such as seismic disturbances from machinery and personnel. Similarly, stage brakes and clamping air-bearings have also proven valuable for improving the stability of certain automated assemblies, and motion-driven structural disturbances and resonances have been pacified by careful motion-profile optimization and by employing nullifying technologies such as Input Shaping®.

All these benefit nano-sensitive applications in many ways, ultimately in terms of yield and throughput. But another, insidious instability process can also bedevil sensitive applications: quasi-asymptotic, nanoscale settling due to lubricant flow processes in drive, alignment and adjustment elements such as leadscrews and micrometers. This is slow, progressing at the nanoscale over many minutes. Since conventional position-metrology instrumentation is typically less stable than recent process tolerances over the timescales involved, quantification of these issues has been lacking.

Here we review:

- An emerging class of motion drive technologies capable of nanoscale resolutions over long travels, including in challenging environments (ultra-clean, high vacuum, fieldless/non-magnetic, and/or EUV);
- A novel metrology approach which allowed, for the first time, comparative quantification of long-term, nanoscale drift/settling behaviors at the nanoscale, and
- A family of actuation mechanisms that mitigates drift issues by eliminating their source.

BACKGROUND

The stability of an apparatus can be no better than the stability of each of its subassemblies. Gradual drift/settling behavior, commencing with motion and perhaps varying in time or with conditions, is particularly problematic to performance, yield and throughput. Some contributors have been analyzed thoroughly in the literature; for example, the dimensional stabilities of structural materials are well established [1].

One contributor that has been analyzed on a material level is the lubricant in the screw mechanism of spindle-driven stages. The lubricant flows and displaces into an eventual equilibrium state. The rheology of lubricants such as greases commonly used in screw actuation mechanisms has been studied extensively, yielding classical models such as the straightforward Bingham Model, supplanted over the years with more elaborate understanding [2] and analyzed experimentally as a function of time in the transition from the dynamic to a quasi-static regime [3].

Unfortunately, understanding of this behavior has remained theoretical due to the instability of conventional position-metrology equipment at the nanoscale over the relevant timescales. It is evident, though, that eliminating such lubricant films in drive mechanisms would improve long-term stability. Applying a stage brake might be another possibility, but these inevitably disturb the position of the mechanism when actuated, and their effectiveness on the nanoscale is unproven for the same reason of inadequately stable metrology over the long term. Replacing the screw with a magnetic linear motor or voice coil would be no answer since those mechanisms hold position by consuming current, contributing to thermal drift and “hunting” behavior by the servo.

EMERGING PIEZO-BASED, LONG-TRAVEL TECHNOLOGIES PROVIDE SELF-LOCKING

Familiar piezo stack actuators, commonly used in nanopositioning components, are composed of layers of ceramic interleaved with electrodes; applied voltages produce controllable dimensional changes which are useful for responsive nanoscale actuation of mechanisms. However, the maximum expansion of these actuators is about 0.1% of their length. Flexure amplifiers have been employed for up to 2 mm of actuation from a stack only a few dozen mm long. Longer travels have traditionally necessitated magnetic motors of various sorts.

That changed with the past few years' innovations in long-travel, direct-drive piezo-ceramic motors. Quite different from stack actuators, these fall into three basic categories, all capable of many centimeters of travel: *walking*, *stick-slip* and *ultrasonic*. Importantly, some configurations engage the ceramic directly with the moving workpiece, such as in the case of an ultrasonic motor whose oscillating frictive tip is preloaded against a drive-strip on the workpiece or the example of a walking actuator whose motive elements are preloaded against a drive shaft. In both cases the screw's lubricant film is eliminated. (Other piezomotor implementations actuate a rotating screw and thus do not eliminate the drift processes under discussion.)

In one family of unlimited-travel technologies based on piezo ceramics that includes resonant [4] and non-resonant [5] approaches, one or more ceramic forcers are preloaded against a ceramic-surfaced stage platen or actuator shaft so that their actuation may confer motion. In some variants, actuation characteristics are similar to a familiar DC servomotor's, in which the velocity is approximately proportional to the magnitude of the applied stimulus. In all cases, the actuation mechanism provides inherent braking force when stimulation is ceased—the mechanisms essentially form a ceramic stage brake.

A direct benefit of these long-travel piezomotors, then, should be improved in-position stability. This has been believed to be the case even down to the nanoscale, although metrology challenges have until recently denied quantitative demonstration.

Resonant Piezomotors

In a specific example of this breed of piezomotors (FIGURE 1), a ceramic wafer is configured so that high-frequency excitation of one of a pair of stimulating electrodes drives one or more resonant modes in its material, conferring a small oscillatory motion to the friction pusher tip affixed at a vibrational antinode for the resonance, typically on the midline of one long side of the ceramic slab. The tip, oscillating like a miniature pogo-stick with nanoscale amplitudes and frequencies of hundreds of kHz, can then be preloaded against a guided workpiece, driving long-travel motion. Actuation speed is approximately proportional to the amplitude of the stimulation, as with a DC servomotor, though (again in analogy to a pogo-stick) a threshold stimulation is required before motion occurs. This deadband behavior is valuable for eliminating hunting behavior by the servo.

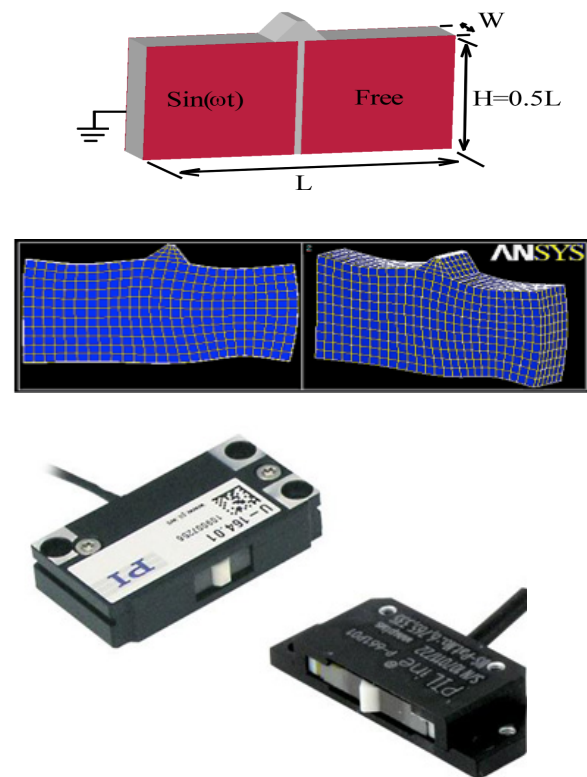


FIGURE 1. Novel resonant piezomotor places a frictive actuating tip at the vibrational antinode of a piezoceramic slab. When preloaded against a workpiece, ultrasonic stimulation of the slab confers motion. When quiescent, it acts as a ceramic stage brake, improving system stability.

By eliminating lead-screws and their inertia and associated linkages and structures, such mechanisms can be very compact and responsive. For example, off-the-shelf linear stages can provide 19mm of travel at up to 500mm/sec and 10gee acceleration with a 0.1 μ m resolution linear encoder, all in a package 35mm square. Even smaller mechanisms have been devised, such as a miniature self-guided piezomotor actuator for miniature camera, fluid delivery and endoscopy applications.

In addition to in-position stability superior to that of conventional stages, other signature advantages of piezo actuation apply such as fieldlessness, vacuum-compatibility, EUV compatibility, and long life.

Piezo Walking Drives

By combining piezo elements acting in longitudinal and transverse directions, the foundation for a walking actuator is composed. These elements can be compressed against a longitudinal rod to confer motion. One familiar progenitor of this family of mechanisms was the Burleigh Inchworm, no longer manufactured. More recently-developed mechanisms offer extraordinary stiffness and holding force and are optimized for reliability in applications requiring long-term position hold while providing centimeters of travel with picometer-class resolution.

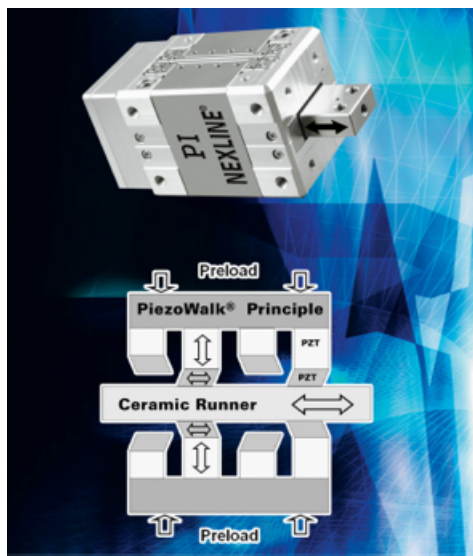


FIGURE 2. Walking motors represent another class of long-travel piezomotors. Composed of forcer elements which directly confer incrementing and fine slew motions to a ceramic

runner, these mechanisms provide cm-scale travel, picometer resolution, long-term position-hold with power off, and actuation forces into hundreds of Newtons.

An even newer solution uses cost-effective bender-type piezo elements. Size and cost are substantially reduced; power-off stiffness is still remarkable, with 10N holding force. This design also provides centimeters of travel range and picometer-class positionability, ideal for applications like optic positioning and sample positioning in e-beam direct-write applications (FIGURE 3).

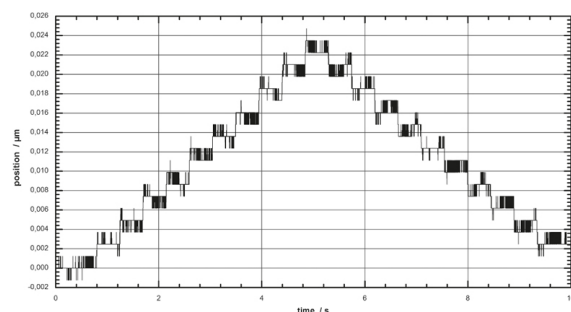


FIGURE 3. Integration of an 0.5nm-resolution encoder in a walking-type piezomotor-driven stage yields these 2nm commanded steps.

In the case of both these walking-style piezomotor types, motion occurs in two modes: a long-travel stepping mode, and a fine-motion analog mode in which the actuator elements are sheared, providing sub-nm positioning capability.

QUANTIFICATION OF LONG-TERM, NANOSCALE POSITION STABILITY

While the stability benefits of piezomotor actuation has been conjectured for decades, documenting stability quantitatively at the nanoscale over long periods has remained an elusive goal. For example, conventional optical position-metrology interferometers have typically lacked the necessary long-term stability to provide quantitative measurement of structural stabilities over periods of many minutes.

Recently, researchers in the field of single-molecule biophysics have utilized optical tweezers in a configuration that quantitatively compared the stabilities of screw-driven and piezomotor-driven stages with sub-nanometer sensitivity over many minutes [6]. As in many semiconductor-industry applications, the test

setup stacks coarse-positioning (resolution 0.1 micron) and fine-positioning motion elements (resolution 0.1 nanometer) in a sophisticated microscopy configuration. Traditionally in this field the coarse-positioning substage has been a screw-driven mechanism, but piezomotor-based microscopy stages have recently been commercialized. By swapping the coarse-positioning stage between screw-driven and piezomotor-driven models, the long-term, nanoscale stability provided by the piezomotors was quantitatively revealed.

The stability of each substage was measured on a dual-beam optical trapping microscope. 600nm-diameter polystyrene beads were adhered to a glass coverslip and mounted on the nanopositioning stage carried by the stacked X-Y substages. The beads' positions were measured by back-focal plane detection [7] using a laser imaged onto a position-sensitive diode (PSD); the combination of clever optics and processing of the PSD signals provided sub-nanoscale localization of the bead in three dimensions. The position of the beads was tracked at five minute intervals, allowing compilation of the long-term overall system drift trends shown in FIGURE 4. The incremental stability provided by elimination of the screw-driven coarse-positioning mechanisms is evident. While long anticipated, such validation of piezomotors' long-term stabilities had not been quantitatively visualized before.

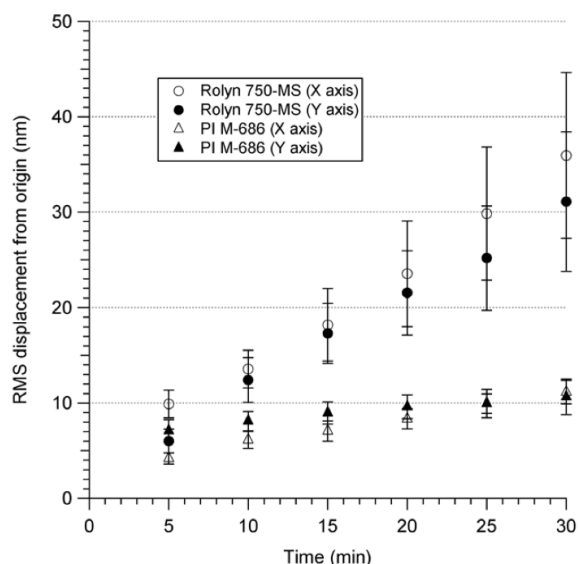


FIGURE 4. From [7], a novel application of optical tweezers allowed long-term, nanoscale quantification of overall system drift in a microscopy configuration including either a

screw-driven coarse-positioning stage or a piezomotor-driven equivalent. As can be seen here, elimination of the actuating screw and its lubricant interface results in significantly improved long-term nanoscale stability, in conformance with theory regarding long-term lubricant flow in the screw assembly.

NEXT: A NEW FRONTIER IN DIMENSIONAL PICOMETROLOGY

Meanwhile the capability of next-generation optical interferometers continues to advance. A key requirement for their industrial deployment is concomitant development of sufficiently accurate metrology and dimensional standards for verification of these next-generation instrumentation and tools. Extraordinary progress has very recently been made in the field of highly accurate dimensional nanometrology by leveraging x-ray diffractive interferometry with recently-developed digital controls approaches. Work performed at European sites including England's National Physical Laboratory have utilized user-programmable FPGA architectures and a novel DAC resolution enhancement technique contributed by the author to servo-control a piezo-driven, monolithic silicon x-ray interferometer, locking it on the diffractive fringe structures. Since the crystallographic lattice structure of silicon is well understood, this approach yields quantized steps of picoscale absolute accuracy. These researchers' demonstration of subatomic absolute accuracies has very recently been published [8]. These instruments are targeted to validate the performance of next generation displacement measuring optical interferometers within the European Metrology Research Program's project NANOTRACE.

APPLICATIONS

Mechanisms potentially benefiting from this capability for long travel with nanoscale resolution and long-term, nanoscale stability range from microlithographic staging and fixture positioning systems to piezomotor variants of classical optic and substrate positioners. The all-ceramic drive principle is amenable to EUV applications and has been implemented in a novel, six-degree-of-freedom, UHV, fieldless, ultra-stable compact hexapod for critical EUV front-end applications.

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ROTATION CONTROL AND DISTURBANCE FORCE COMPENSATION OF AIR TURBINE SPINDLE FOR PRECISION MACHINING USING HIGH-PRECISION, QUICK-RESPONSE PNEUMATIC PRESSURE REGULATOR

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INTRODUCTION

This develops a quick and robust rotation control method for air turbine spindles used in ultra-precision machining. Air turbine spindles with aerostatic bearings are widely used in ultra-precision machines such as aspherical lens generators [1] due to their beneficial properties, which include high precision, high speed, low friction, and low vibration. However, air turbine spindles currently suffer from the following problems:

- 1) The rotation speed of the spindle tends to decrease when a disturbance force (e.g., cutting force) is imparted to the spindle.
- 2) The spindle rotation speed changes when the air pressure supplied by the compressor changes.

Some studies have reported the relationships among the supply pressure, the flow rate, the rotation speed, and the output torque of the air turbine spindle. However, because methods to control the air supply pressure and the flow rate are too imprecise and slow, it has not been possible to control the rotation speed and the output torque of an air turbine spindle.

To address this problem, we have developed a high-precision, quick-response pneumatic pressure regulator (HPQR) [2] and applied it to rotation feedback control of an air turbine spindle.

FABRICATON OF HPQR

HPQR Configuration

Fig. 1 shows a photograph of a fabricated HPQR. The HPQR is a pressure regulator that can control the supply pressure very quickly and precisely. It consists of a spool-type servo valve (SP valve; Festo, MPYE-5-1/8LF-010B), an isothermal chamber, a pressure differentiator (PD sensor; Tokyo Meter, DTP-8), a pressure

sensor (Toyoda, PD-64S500K), and a DSP (Digital Signal Processor) controller (MTT, s-BOX). The SP valve has five ports, but it is used as a three-port (namely, supply, control, and exhaust ports) servo valve; the two unused ports were plugged. The isothermal chamber is a chamber filled with metal wool, which creates a nearly isothermal state in the chamber. The isothermal chamber has a volume V of 0.001 m^3 . The PD sensor measures the pneumatic pressure differential with high precision and a quick response.

Controller Design

The purpose of HPQR is to maintain the pressure in the isothermal chamber at a set value. The important point is that, irrespective of upstream or downstream disturbances, a

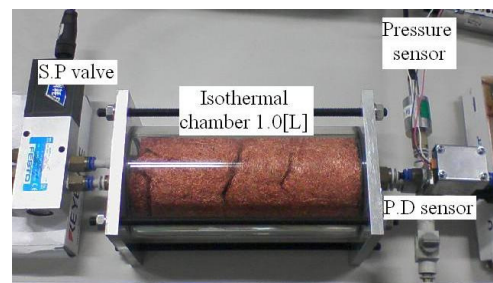


Figure 1: Photograph of the HPQR

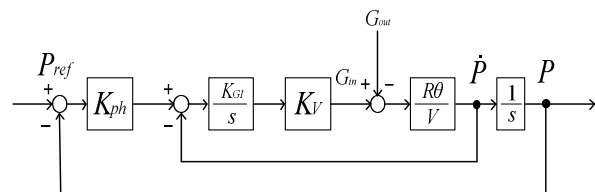


Figure 2: Block diagram of proposed pressure regulator

constant pressure should be maintained in the isothermal chamber. Fig. 2 shows a block diagram of the developed HPQR. The symbols in Fig. 2 are defined as follows: P_{ref} : reference pressure [Pa], K_{ph} : proportional gain [(kg/s)/Pa], K_{Gi} : integral gain [V/kg], K_v : flow rate gain of SP valve [(kg/s)/V], G_{in} : input flow rate [kg/s], G_{out} : output flow rate [kg/s], R : gas constant [J/(kg K)], θ : chamber temperature [K], V : chamber volume [m³], P : pressure in chamber [Pa].

The SP valve characteristics are approximated by the linear constant K_v . The main loop is a pressure feedback loop. The differentiated pressure is a feedback parameter in the minor loop. Thus, a PID control system is used to maintain a constant pressure in the isothermal chamber, irrespective of upstream or downstream disturbances. The minor loop also reduces the effects of the nonlinear characteristics of the SP valve and supply pressure fluctuations. Assuming that the minor loop has much shorter time constant than the main loop, the transfer function of the main loop can be expressed by

$$\frac{P}{P_{ref}} = \frac{1}{\frac{K_{ih}K_v}{K_{ph}}s^2 + \frac{1}{K_{ph}}s + 1} \doteq \frac{1}{1 + T_p s} \quad (1)$$

In this study, K_{ph} was set to 20 (kg/s)/Pa, so T_p was calculated to be 0.05.

ROTATION FEEDBACK CONTROL OF AIR TURBINE SPINDLE WITH DISTURBANCE FORCE OBSERVER

Feedback Control of Rotation

Figs. 3 and 4 respectively show the configuration and a block diagram of the rotation feedback control system of the air turbine spindle (NSK Nakanishi, ABS-1600). As shown in Fig. 3, two air supply ports are used with the aerostatic bearing air turbine spindle: one is for the aerostatic bearing and the other is for driving the air turbine spindle. Only the pressure for the air turbine spindle was controlled by the HPQR, whereas the pressure for the aerostatic bearing was controlled by a commercially available diaphragm pressure regulator. The rotation speed is measured by a photo IC (Ono Sokki, FG-1200) and the signal is fed back to the DSP. As shown in Fig. 4, a PI controller is used in this preliminary experiment and the dynamic characteristics of the air turbine spindle are modeled as a first-order lag system ($T_i = 3.3$ s; $\alpha = 0.5$ min⁻¹/Pa). The PI controller gains are set

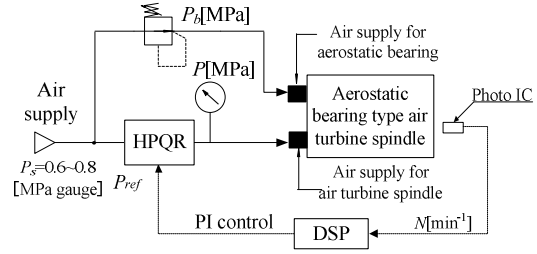


Figure 3: Configuration of experimental apparatus

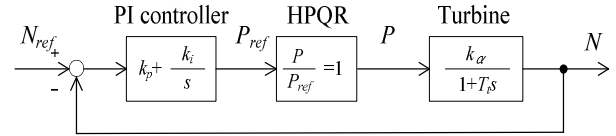


Figure 4: Block diagram of rotation feedback control

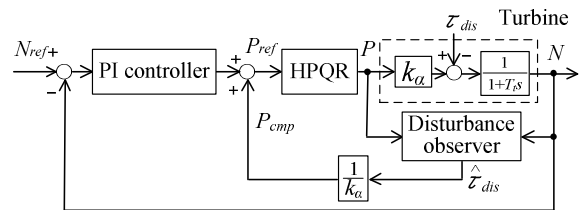


Figure 5: Block diagram of rotation feedback with disturbance observer

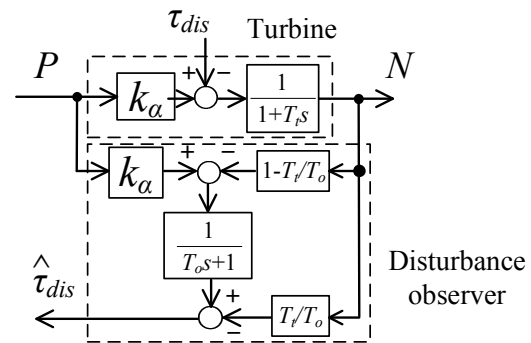


Figure 6: Block diagram of disturbance observer

so that the time constant of rotation control is 0.5 s.

Disturbance force observer

A force disturbance observer [3] is added to the controller shown in Fig. 4 to compensate the rotation change caused by the cutting force disturbance. Figs. 5 and 6 respectively show

overview and detailed block diagrams of the disturbance observer. This observer estimates the cutting force disturbance τ_{dis} imparted to the edge of spindle (tool) from the measured pressure P [Pa] and the rotation speed N [rpm]. The observer filter is set to $T_0 = 0.7$ s. When a disturbance force τ_{dis} is imparted to the spindle, the set pressure value P_{cmp} is immediately added to the reference value of the HPQR to generate the compensation force.

DIAMOND CUTTING EXPERIMENT USING THE DEVELOPED SYSTEM

Rotation Control Experiment

Before applying the proposed fabricated air turbine spindle control method to a diamond cutting experiment, a rotation control experiment was conducted to confirm the performance and stability of the rotation control. The experimental configuration was the same as that shown in Fig. 3. Table 1 lists the specifications of the air turbine and the force sensor.

In this experiment, the target value of the rotation N_{ref} was first set to $N_{ref} = 40,000 \text{ min}^{-1}$; this value was then changed to $80,000 \text{ min}^{-1}$ at $t = 100$ s. Fig. 7 shows the experimental results. The broken blue line in Fig. 7 shows another experimental result for comparison. In the preliminary experiment, the static characteristics of the air turbine spindle between the supply pressure P and the rotation N was measured. Using the static experimental result as a reference, the pressure was controlled in a step like manner using the HPQR (initially 47.0 kPa and then 107.5 kPa after $t = 100$ s, without a rotation speed feedback loop). The broken blue line in Fig. 7 indicates the result obtained. In Fig. 7, the time constant is approximately 0.5 s with the proposed rotation speed feedback system, whereas it is approximately 3.3 s without rotation speed feedback (only pressure control). The maximum error from the target value is 30 min^{-1}

Table 1 Experimental specifications

Workpiece	Size	20×20×5 [mm]
SUS430	Axial depth of cut	0.8 [mm]
	Radial depth of cut	0.05 [mm]
Air turbine spindle	Tool rotation	60,000 [min^{-1}]
NAKANISHI	Turbine pressure	0.071 [MPa]
ABS-1600	Air bearing pressure	0.5[MPa]
X, Y, Z-stage	Max. displacement	85 mm(X, Y), 35 mm(Z)
SIGMA KOKI	Displacement of each pulse	0.001 [mm]
SGSP20-85,35	Cutting speed	1[mm/s]
Force sensor	Range	10 [N]
	Threshold	0.01 [N]
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with the proposed system, whereas the maximum error is approximately $4,000 \text{ min}^{-1}$ without the rotation speed feedback. This demonstrates the effectiveness of the proposed system.

Diamond cutting experiment

We apply the proposed fabricated air turbine spindle control method to diamond cutting of a stainless-steel alloy. Fig. 8 shows the experimental apparatus. Table 1 lists the specifications of the experimental apparatus and conditions. In Fig. 8, the air turbine spindle with a square carbide end mill (two teeth; 0.8 mm in diameter) is fixed horizontally. A stainless-steel alloy (SUS430) workpiece is set on an X–Y positioning stage. A force sensor between the vice and the workpiece is used to measure the cutting force in the X direction. In the experiment, N_{ref} was first set to $N_{ref} = 60,000 \text{ min}^{-1}$; the experiment started after the rotations had settled. Fig. 9 shows the experimental results. In Fig. 9, the cutting experiment begins at $t = 5$ s and ends at $t = 25$ s. The green line indicates the variation in the cutting force in the X direction measured by the force sensor. The red line indicates the experimental result for rotation for the proposed method. Fig. 10 shows an enlarged graph. The broken blue line in Fig. 9 indicates the result obtained using ordinary pressure (rotation) adjusting method (supply pressure P is adjusted by a commercially available diaphragm pressure regulator at 0.071 MPa, without a rotation speed feedback loop). The experimental results clearly show that the rotation value obtained using the conventional method (diaphragm regulator) drops to around $53,000 \text{ min}^{-1}$, whereas for the proposed method, the rotation changes by at most 100 min^{-1} and then immediately recovers.

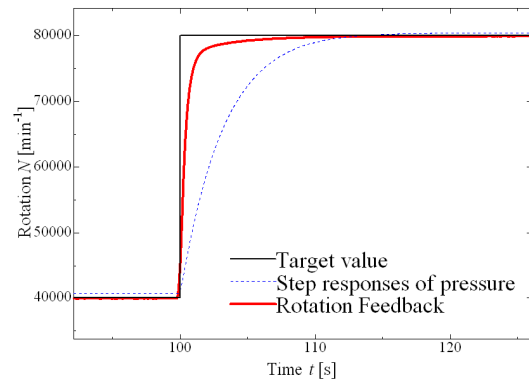


Figure 7: Experimental results (step responses of spindle revolution)

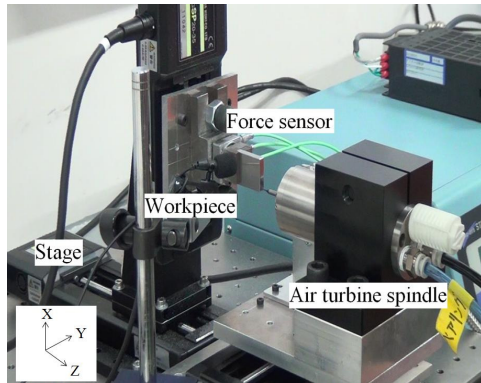


Figure 8: Diamond cutting experiment

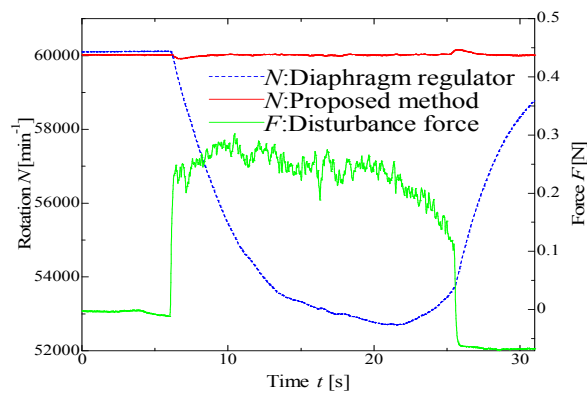


Figure 9: Diamond cutting experiment

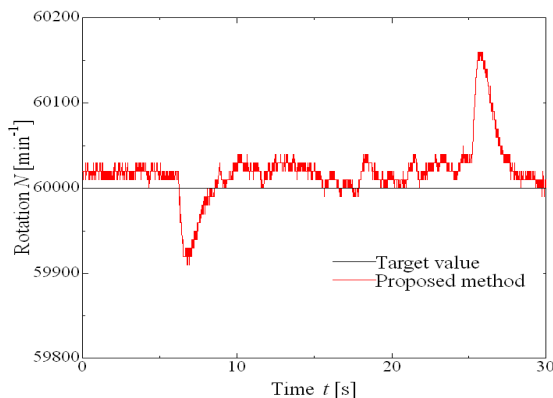


Figure 10: Diamond cutting experiment

CONCLUSION

We have developed a high-precision, quick-response pneumatic pressure regulator and applied it to rotation feedback control of an air turbine spindle. We also set a disturbance force observer in the feedback control system to avoid changes to the rotation speed due to disturbance forces imparted to the air turbine spindle.

The effectiveness of the proposed method was experimentally demonstrated through a rotation control experiment and a diamond cutting experiment.

ACKNOWLEDGEMENT

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ANALYSIS OF THE EFFECT OF PWM ON POSITION CONTROL USING DYNAMIC ERROR BUDGETING

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INTRODUCTION

Industry is continually demanding machines that are increasingly faster, more accurate and less expensive. This presents designers with the challenge of being able to produce machines that they feel confident can meet industry's needs. The high cost of building a machine requires designers to know ahead of time the amount of error one can expect. The technique known as dynamic error budgeting allows a designer to look at a machine in terms of subsystems, and consider how much error can come from a number of error sources as a function of frequency [1]. It allows disturbances and noise to be modeled as a power spectral density in the frequency domain and with a system model, each subsystem's effect on the output can be determined.

This type of analysis has its roots in noise analysis. Early work with power spectral densities was done to analyze noise in communication signals. A signal was passed through a number of band pass filters, the outputs of which were each sent through a resistor. Calorimeters were used to measure the thermal power output by each of the resistors. These power measurements could be plotted to produce the power spectral density of the signal [2]. Since then, the idea of a power spectral density to analyze the frequency content of noise or error has been used in a variety of applications. Gans has used it to study errors in radio communication [3] while Abramovitch used it to characterize the error in a disk drive [4].

* * *

Despite the rise in popularity of linear voltage amplifiers in order to control precision positioning machines, many high torque applications still necessitate the usage of pulse-width modulated (PWM) amplifiers to power their systems. The constant switching inherent in these amplifiers introduces errors into the

system. However, there is currently no documented way of quantifying the amount of noise that this creates in the position output of the system.

Following a brief summary of dynamic error budgeting, this paper examines the effect of PWM switching on the position control of a precision machine using the framework of dynamic error budgeting. The idea will be derived and demonstrated for the case of a simple DC motor.

AN OVERVIEW OF DYNAMIC ERROR BUDGETING

In the design of precision machines, an error budget is often used to determine the total amount of error that is expected from a machine by combining the errors from a list of known error sources (eg. volumetric errors, thermal errors, servo errors, etc.). However, it is often useful to describe an error source as a function of frequency, or as a power spectral density. This way, the error source can be considered together with all of the system's dynamic interconnections, thus taking into effect its various closed loop transfer functions, to get the power spectral density of the system's output. It allows the designer to consider the performance of the system as a function of frequency (i.e. dynamically) and not just statically. In addition to understanding how frequency affects the performance of a system, it allows the designer to determine an expected variance in an output of concern, as a result of each of the various noise sources and disturbances.

In general, the process of dynamic error budgeting is as follows. Once a system is designed, its mechanical components are modeled so that when combined with the control elements, one can obtain a full frequency domain model of the control loop. Then, all of the outputs are identified, along with all the noise sources or inputs that affect them. The noise sources must then be modeled as power

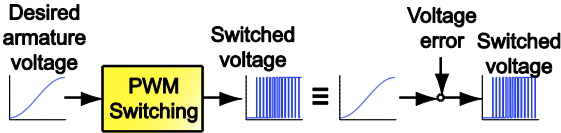


FIGURE 1. PWM switching can be replaced by an equivalent noise injection.

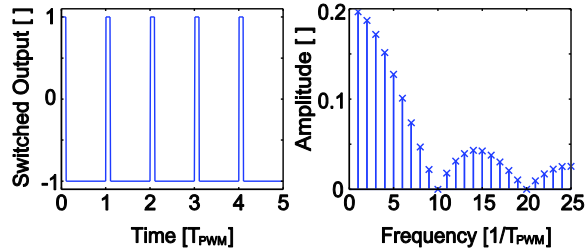


FIGURE 2. The first 50 harmonics of a PWM switched voltage error signal at $\alpha = 0.05$. Note that the axes are normalized for a unit output.

spectral densities. The transfer matrix from each of the noise inputs to the performance outputs can then be obtained. With this transfer matrix, $\mathbf{H}$, and the vector of power spectral density of the noise inputs, $\mathbf{S}_d$, the vector of the power spectral densities of the outputs, $\mathbf{S}_x$, can be obtained as follows [1]:

$$\mathbf{S}_x(f) = \mathbf{H}(j2\pi f) \mathbf{S}_d(f) \mathbf{H}^T(j2\pi f) \quad (1)$$

A cumulative sum of $\mathbf{S}_x$ over all frequencies up to the system's Nyquist frequency can then be taken in order to get a vector of the outputs' cumulative power spectra. They can also be summed outright to get the variance of each output. Plotting the power spectral density of the outputs tends to show peaks, each of which can typically be attributed to the frequency content of one of the noise inputs. Plotting the cumulative power spectrum allows the designer to see the contribution of each of the peaks to the overall variance of the output. It should be noted that the final (high frequency) value of the cumulative power spectrum is equal to the predicted variance of the output signal

A more in depth description of dynamic error budgeting can be found in [1] and [5].

PWM AS A POWER SPECTRUM

In order to study the effects of PWM switching, first the voltage errors caused by switching must be modeled as a disturbance. Although in reality the amplifier produces a signal that alternates between its positive and negative voltage rails, it

is more convenient, and necessary for the purposes of dynamic error budgeting, to consider the equivalent system where we add an error to a linearly applied potential. This error is the difference between the desired voltage and the switched signal (Fig. 1). The voltage error signal resembles a square wave with zero mean and with the same duty cycle as the PWM signal. This way of modeling the PWM effectively creates a new input into the system which can be conveniently characterized. It is possible to derive a Fourier series (Fig. 2) that describes the frequency content of this signal. For a PWM waveform with amplitude A , period T_{pwm} , and duty cycle α ($0 \leq \alpha \leq 1$), the voltage, V , can be shown to be equal to the following Fourier series representation:

$$V(t) = \sum_{n=-\infty}^{\infty} D_n e^{jn\omega t} \quad (2)$$

$$D_n = \frac{Aj}{\pi n} (e^{-jn\alpha 2\pi} - 1) \quad (3)$$

The power spectral density of the voltage error caused by the PWM switching, S_v , can then be obtained by taking the square of the magnitudes of the Fourier coefficients, D_n .

$$S_v(f) = \frac{2A^2}{\pi^2 T_{pwm}^2 f^2} (1 - \cos[2\pi T_{pwm} f \alpha]) \quad (4)$$

Once the power spectral density of the voltage error is known, it can be used to calculate the power spectral density of the error it causes in the output of the system; in most cases, position error.

VALIDATION IN SIMULATION

In order to validate the proposed extension of dynamic error budgeting, control of a DC motor is simulated in which the motor voltage is controlled with a PWM amplifier. In this simulation, a motor under PID position control and PI current control are considered. The parameters of the drive system have been summarized in Table 1. The digital control sampling frequency and PWM frequency are both 10 kHz. The conceptual block diagram of the system is shown in Fig. 3. The dynamic error budget was produced according to [1].

With the drive holding a constant position, with no disturbances acting on it other than the switching of the amplifier, the motor armature should see no voltage or current. As such, for

TABLE 1. List of control system parameters.

Physical Constants			Current Controller		Position Control		
M_{eg} [m/s ² A]	R [Ω]	L [H]	K_i [V/As]	K_p [V/A]	K_i [A/ms]	K_p [A/ms]	K_d [As/m]
0.698	0.12	0.001	1.6e4	5.7	1.5e7	1.5e5	261

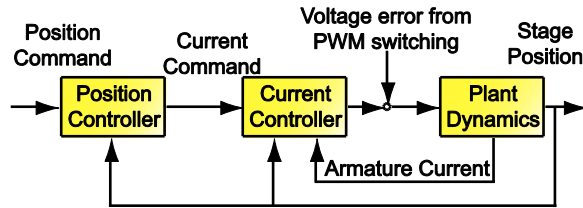


FIGURE 3. Conceptual block diagram showing how PWM error is injected into the system.

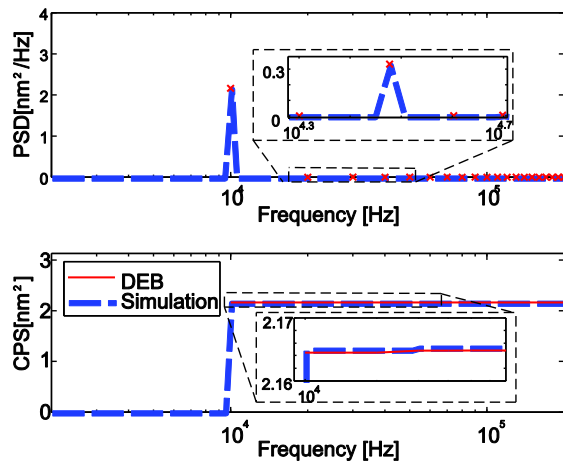


FIGURE 5. Power and cumulative power spectra predicted by the dynamic error budget and validated in simulation for a single mass system with a PWM amplifier acting at 10 kHz with an average duty cycle of 50%.

this study, a value of 0.5 is selected for the duty cycle, α , resulting in a zero mean input voltage. The power spectral density of the voltage error can then be calculated as detailed earlier on.

Since in this case, the system has only one input: the voltage error from the PWM and one output: the drive position, the power spectral density of the output can be obtained by simply multiplying the power spectral density of the PWM disturbance with the square of the magnitude of the transfer function from the PWM voltage error to the position output of the drive. The result is a power spectral density that describes the error in position caused by PWM switching. The expected variance of the time domain position output is obtained by summing up this power spectrum across frequency. The

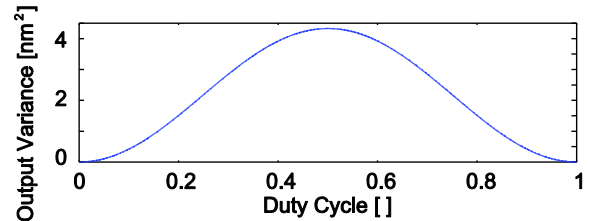


FIGURE 4. Plot showing the effect of PWM duty cycle on position variance.

predicted power spectral density (PSD) and cumulative power spectrum (CPS) are shown with the continuous line in Fig. 4. We can see that the majority of the variance in the output is caused by the step-up at the PWM frequency of 10 kHz. In addition, the cumulative power spectrum keeps creeping up at higher frequency due to higher harmonic components of PWM switching.

The error budget was validated by simulating a system with an actual PWM switching block. The power spectral density of the drive position was obtained by taking a fast Fourier transform of the time domain position signal, and then taking the square of these amplitudes. The simulation result is shown with the dashed thick line in Fig. 4.

It should be noted that as there are no disturbances or additional servo errors in the system, the duty cycle of the voltage signal is 50% (i.e. switching between maximum and minimum line voltages). If one forces a different duty cycle by injecting a constant force disturbance, the variance of the output changes according to the pattern shown in Fig. 5. This change in variance can be attributed to changes in the PWM voltage error harmonics. It can also be seen that as far as PWM induced servo error is concerned, regulation around zero load (and zero current) seems to create the worst case situation.

CONCLUSION

The scope of dynamic error budgeting has been expanded to include the modeling of the effect of PWM switching. One apparent issue is that the frequency content of the equivalent voltage error, used to capture the effect of PWM, depends on the commanded duty cycle. Thus, separate dynamic error budgets must be conducted at a set of expected operating conditions for the actuator(s). The frequency domain predictions closely match the simulation results obtained by considering the time-domain

behavior of a PWM controlled drive. Future work will focus on experimental validation of this model and extension of the methodology to 3-phase permanent magnet synchronous motors, which are used commonly in industry.

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MODEL BASED CALCULATION OF THE TOOL-PATH SMOOTHING TOLERANCES

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INTRODUCTION

Smoothing the tool path and the tool orientation within a certain tolerance is a common approach for reducing the mechanical excitation of a machine tool [1]. This is normally done by insertion of C_n -continuous polynomial-functions like NURBS or B-Splines assuming a constant value for the smoothing tolerance along the whole tool path. This value has to be smaller than the final tolerance value which is obtained for the finished part because of additional machining errors during the manufacturing process. These errors consist of geometric, dynamic and thermal errors as the effect of follow-up errors of the axis drives [2]. Non-regarding these errors causes a very small and constant value for the smoothing tolerance. This leads to higher curvatures of the tool path and finally to a higher mechanical excitation of the machine tool for a given path velocity. Figure1 illustrates the effect of different path tolerances on the resulting curvature for corners having an angular range between 0 (straight) and 120 degrees.

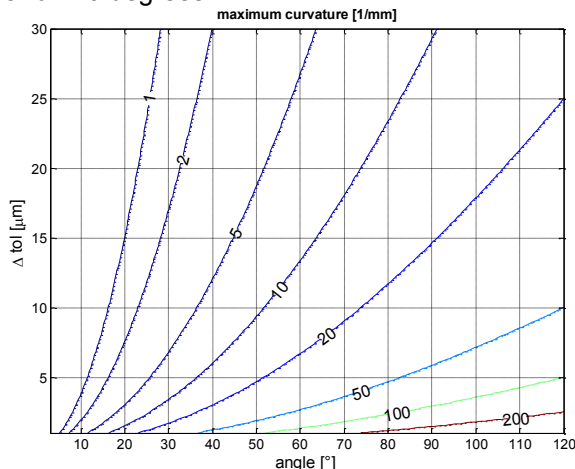


FIGURE1. Resulting curvatures for a smoothed corner depending on the smoothing tolerance and the angular range of the corner.

There are several ways known for measuring and modeling the machining errors in order to

compensate them [3]. Regarding the smoothing of the tool path and the tool orientation current smoothing-algorithms are not able to predict the diversity of the machining uncertainties along the tool path. Here, a simple tolerance model for the prediction of geometric uncertainties along the tool path for a given work piece and a given machine is presented and applied for the two and five axis case.

At first, the geometrical errors of the linear axes and the resulting uncertainties for 3-axis machine tool movements are shown. Next, the geometric uncertainties of rotational axes and their contributions to the resulting uncertainties at the Tool-Center-Point (TCP) are added. In a final step the calculation results of the available smoothing tolerance are shown.

GEOMETRICAL UNCERTAINTIES OF A 3-AXIS MACHINE TOOL

In this section the calculation and merging of geometric uncertainties is shown for the example of a movement of 3 cartesian axes. As defined in ISO 230-1 [4] there are 6 element deviations defined for each linear axis listed for the example of a linear X-axis:

- EXX: positional deviation
- EYX: linear deviation
- EZX: linear deviation
- EAX: roll deviation
- EBX: pitch deviation
- ECX: yaw deviation

The resulting 18 element errors for 3 linear axes and the 3 squareness errors

- C0Y: out of squareness between X and Y
- B0Z: out of squareness between X and Z
- A0Z: out of squareness between Y and Z

yield to 21 geometric errors for a 3 axis machine. All these error contributors are merged to the axis-wise geometric standard uncertainties U_x , U_y and U_z , so by knowing the geometric errors, the tool path and the part dimensions for every position of the trajectory the geometric uncertainties can be calculated. Depending on

the regarded machining process there exists a relevant direction for the uncertainty. For the example of 2D laser cutting Figure2 shows the effective uncertainty in lateral direction for a 2D work piece. In this case, for a quite small work piece and when only two axes are involved in the movement, the resulting uncertainties only slightly differ along the path.

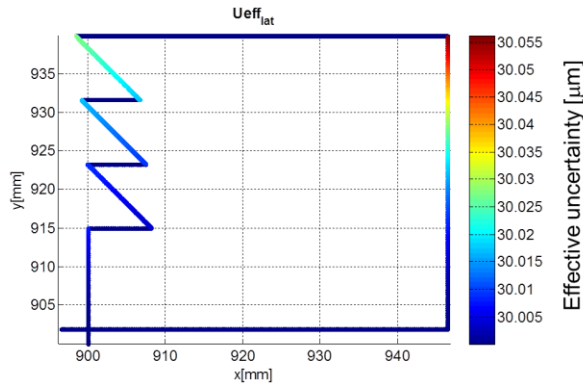


FIGURE2. Effective uncertainty in lateral direction for a 2D laser cutting work piece

GEOMETRICAL UNCERTAINTIES OF A 5-AXIS MACHINE TOOL

Regarding a rotational axis there are 6 element deviations defined, listed for the example of a rotational C-axis:

- EXC: radial error motion
- EYC: radial error motion
- EZC: axial error motion
- EAC: tilt error motion
- EBC: tilt error motion
- ECC: positional deviation

Together with the geometric errors of the 3 linear axes, the component errors for 2 rotational axes and the additional location errors lead to the geometric errors for a 5 axis machine tool movement.

The resulting combined uncertainties are calculated as follows: The geometry of the tool consisting of translatoric and rotatoric set-point trajectories also including the orientation of the tool is defined at the TCP. After transforming the TCP into the Machine-Coordinate-System (MCS) the movements of the machine tool axes are known. After identifying the geometric errors for each axis [5] the geometric uncertainties can be calculated and merged as combined standard deviations for the TCP movement for each cartesian direction at the TCP.

For the example of laser cutting Figure3 shows a typical 3D-movement including changes of the orientation of the tool along the TCP-path. This

leads to large compensating movements of the linear axes which is shown in Figure2.

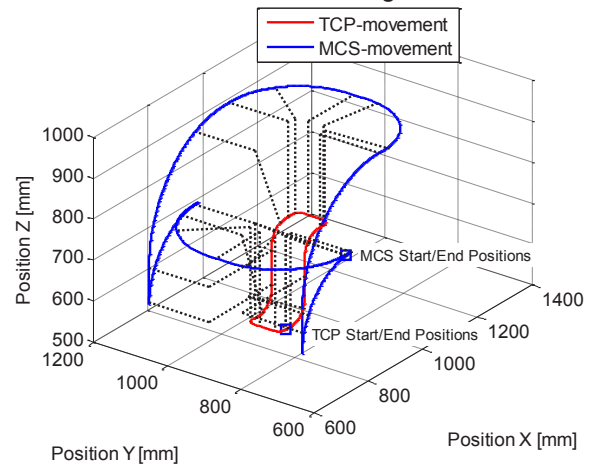


FIGURE3. 5-Axis movement shown for 5-axis laser-cutting.

Due to the increased number of axes, but also caused by the large compensating movements the uncertainties strongly increase. The deviations in x and y -direction which finally lead to the effective uncertainty in lateral direction are shown in Figure4-5. The deviation in z-direction has no influence on the effective uncertainty in lateral direction and is not regarded in this example. EXCTCP, EYCTCP, EX0XY and EY0XY in Figure4-5 are contributors caused by element and out of squareness errors of the involved rotational axes. Because of the large distance between the TCP and the rotation-center of these axes the resulting TCP uncertainties of these contributors are dominant.

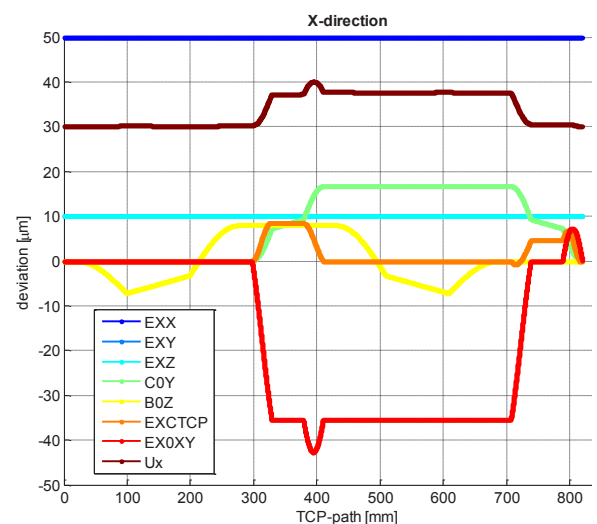


FIGURE4. TCP-deviations in x-direction along the TCP-path

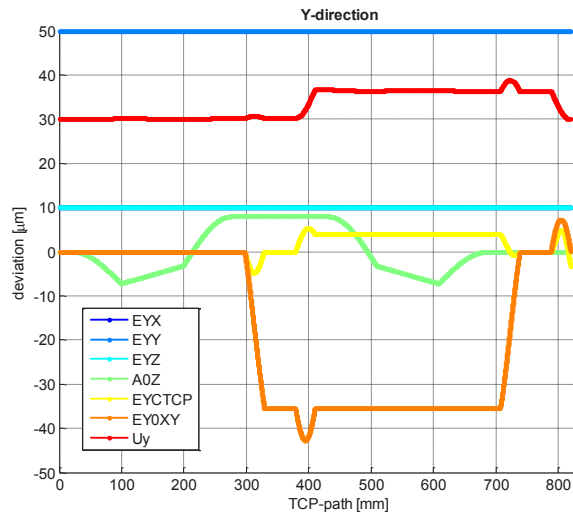


FIGURE5. TCP-deviations in *y*-direction along the TCP-path

The resulting effective uncertainty of the TCP in lateral direction is shown in Figure6.

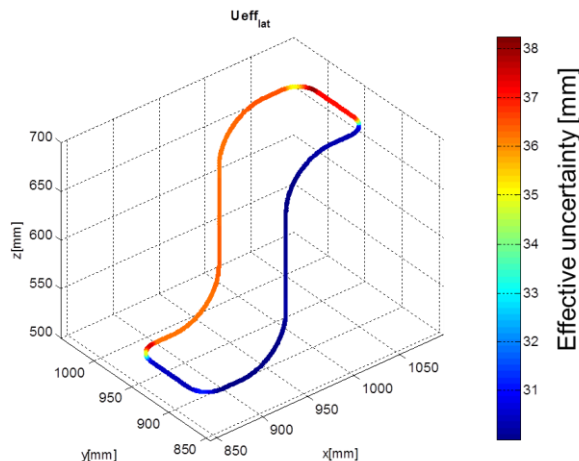


FIGURE6. Effective TCP-uncertainties for a 5-Axis movement shown for 5-axis laser-cutting.

CALCULATION OF THE AVAILABLE SMOOTHING TOLERANCE

The following 3 steps finally lead to the available smoothing tolerance:

1. Calculation of the so-called NC-tolerance based on the technical drawing and the tolerance value which is obtained for the finished part assuming an ideal machine tool with no deviations.
2. Calculation of the effective uncertainties based on component and location errors as illustrated before.
3. Reduction of the NC-tolerance calculated in step1 by

- constant values for
 - thermal deviation
 - dynamic deviation
 - process deviation
- position dependent values for the geometric uncertainties

CONCLUSION

A procedure for the elaboration of the available smoothing tolerances based on the geometric errors of a machine-tool has been shown for the two and five axis case. Depending on part geometry and the allowable range for application of path smoothing can be determined.

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RETROFITTING A PARKER LIBERTY ONE CYLINDRICAL GRINDER WITH AIR BEARING SPINDLES

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ABSTRACT

Air bearing spindles are not standard items on commercially available cylindrical grinders, but they are useful for achieving single digit micro-inch roundness and straightness accuracies. Since our early days, Professional Instruments Co has retrofit our spindles on to vintage hand fed machine tools, but recently we have started buying new CNC machines without any spindles installed. This work describes one of those projects.



FIGURE 1. Parker grinder purchased without spindles.

BACKGROUND

Machine tool builders in general and CNC grinder manufacturers in particular have focused on control and positioning issues more than on spindle and work holding. This is understandable given what their customers are willing to spend on machine tools, and the fact that most grinding shops are working to “tenths”, not millionths.

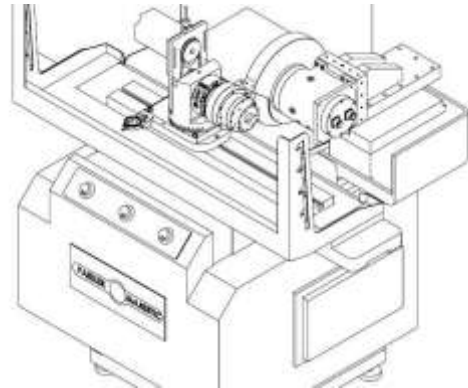


FIGURE 2. Schematic showing spindle placement.

RETROFIT

We start with a Parker Liberty One CNC cylindrical grinder, which we purchased without spindles and add two 4R Blockhead air bearing spindles. The work spindle is belt-driven by the motor provided by the manufacturer and the wheel is mounted on one of our “Twin-Mount” brushless DC spindle packages.



FIGURE 3. Completed spindle retrofit.



FIGURE 4. Grinding an encoder mount.

RESULTS

This is the 4th Parker Liberty we've purchased and retrofit with our air bearing spindles. We put this machine into production in December 2011 and have achieved better than 5 micro-inch roundness.



FIGURE 5. Roundness testing set up.

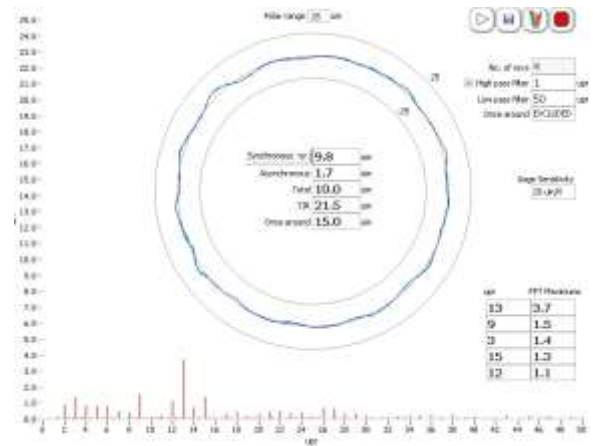


FIGURE 6. Typical results, better than 10 microinch roundness.

CONCLUSIONS

Grinding air bearing components to sub 1/4-Micron tolerances is challenging. Commercially available CNC grinders are capable of hitting these tolerances, but in general the spindles they come with are not quite up to the task. We have found that retrofitting CNC grinders with our standard 4R Blockhead air bearing spindles is an effective and economical way to get around this problem.

IDENTIFICATION OF THE INFLUENCE OF THE DIFFERENT AIR BEARINGS COMPRISED IN AN AXIS OF ROTATION SYSTEM ON THE RADIAL ERROR MOTION

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ABSTRACT

Current trends in high-precision applications stress the need for reliable and well-designed air-bearing systems. In this work, the influence and the interaction are studied of the different air bearings of an axis of rotation system on the radial error motion with the objective to increase the running accuracy to the sub-nanometer level (< 5 nm). To this end, a 5-DOF film model is developed that can account for the minutest details affecting the error motion and the dynamic film forces.

INTRODUCTION

The running accuracy, i.e. radial error motion, of a well-designed air bearing will be mainly determined by the machining accuracy of the bearing surfaces. This is so because the clearances of air bearings should be made as small as possible in order to obtain high stiffness. However, when production capabilities reach their limitations, and so threaten to limit the attainable running accuracy of air bearings, other solutions have to be devised in order to ensure that air bearings can still reach the accuracy needed for ultra-precision applications in an affordable way. This, however, can be achieved by mitigating and/or compensating for the influence of the form errors of the component parts by optimizing the design parameters of an aerostatic bearing.

In a previous work [1] [2], the radial error motion of an aerostatic journal bearing was studied and optimized. In this work, the influence and the interaction are studied of the different air bearings constituting an axis of rotation system on the radial error motion with the objective to increase the running accuracy.

PROBLEM STATEMENT

An in-house developed finite-difference model of gas film behavior was adapted in order to analyse the influence of various manufacturing errors and feedhole configurations on the radial er-

ror motion of an aerostatic journal bearing (2-DOF film model) [1]. The influence of several bearing parameters on the running accuracy is analyzed with the aid of this reliable numerical model [2]. Results show that the radial error motion of an aerostatic journal bearing can be reduced most effectively by increasing the number of feedholes N_f as is illustrated in Fig. 1.

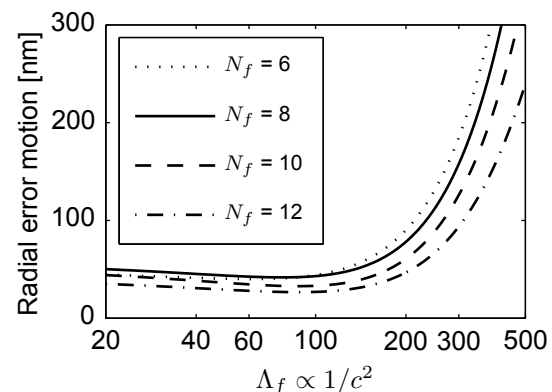


FIGURE 1. The radial error motion as function of the feed number Λ_f evaluated for a various number of feedholes N_f .

From this figure it is clear that the radial error motion will decrease by increasing the number of feedholes, on the one hand, and by selecting the optimal clearance c on the other hand, which coincides with the clearance that yields maximum stiffness as illustrated in Fig. 2.

However, the number of feedholes of an air bearing rotary system is restricted, from practical point of view. A porous material on the other hand has an infinite number of feedholes (ideally). For this purpose, the finite difference model of gas film behavior was adapted and validated theoretically in order to analyse the radial error motion of a porous aerostatic journal bearing. The first results were very encouraging. Analyses showed a sub-nanometer radial error motion for an ideal

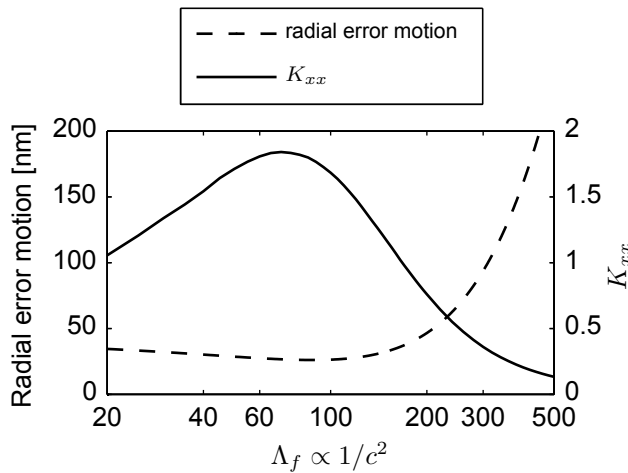


FIGURE 2. Influence of the feed number Λ_f on the radial error motion and the dimensionless stiffness K_{xx} .

porous aerostatic journal bearing with a diameter of 50 mm.

Axis of rotation system

As depicted in Fig. 3, an axis of rotation system comprises typically a journal bearing and two thrust bearings. In order to validate the results of the porous model, an existing aerostatic rotary table was adapted to a porous aerostatic journal bearing. However, the thrust bearings, each made up of eight inherent restrictors, were not adapted. The aerostatic rotary table is shown in Fig. 4. It is placed on a measurement setup used to remove the artifact form error from the radial error motion using a reversal technique [3]. The radial error motion is measured 95 mm above the center of mass (COM) using a single capacitive probe¹. The error motion is measured three consecutive times under three different orientations by rotating the whole setup using an indexation table. This overcomes the need for accurate indexing of the artifact and sensor(s) during testing, as well as the effects of varying sensor sensitivities when multiple sensors are used. The least-squares synchronous radial error motion of the rotary table under test, turns out to be 8 nanometer as illustrated in the polar plot of Fig. 5. This result clearly differs from the sub-nanometer radial error motion calculated with our 2-DOF model. The following argumentations show that this difference can presumably be attributed to the tilt

¹The COM lies above the flange of the rotor because of the mass of the top plate and the artifact, which are mounted on top of the rotor as is visible in Fig. 4.

error originating from the thrust bearings, which are not taken into account in our 2-DOF model:

- The amplitude of the harmonics $n = k \cdot N_f \pm 1$ with $k \in \mathbb{N}^*$ in Fig. 5 (filled circle) is remarkably higher than the amplitude of the remaining harmonic components (circle).
- The axis-of-rotation error reduces when the influence of the thrust bearings is reduced. To this end, the top-thrust bearing was removed from the test setup, while the clearance between the flange of the rotor and the bottom-thrust bearing was increased by increasing the supply pressure. The result of this experiment is depicted in Fig. 6. It is clear that the overall amplitude of the harmonics n decreases as the clearance h of the thrust bearing increases. This result indicates once more that the high amplitudes of the harmonics n originate from the tilt error of the thrust bearings.

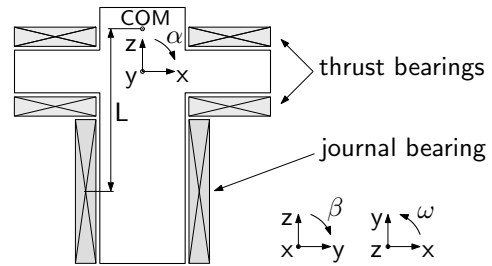


FIGURE 3. Rotor-bearing configuration of the axis of rotation system under study.

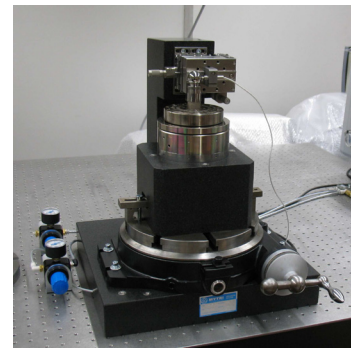


FIGURE 4. Measurement setup used for separating the artifact form error from the radial error motion.

As a result, the influence of the thrust bearings cannot be ignored in our model. To account for this, in this study, a coupled film model is de-

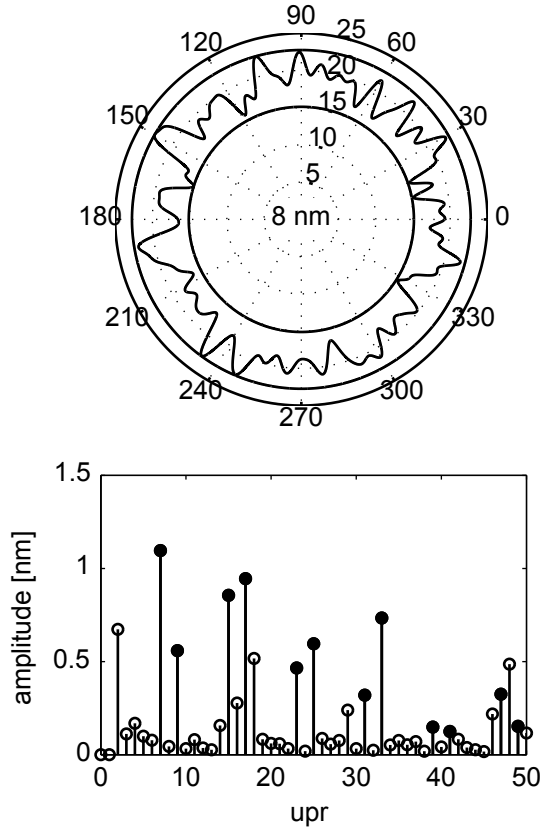


FIGURE 5. Polar plot (top) and spectrum (bottom) of the measured synchronous radial error motion of the rotary table under test.

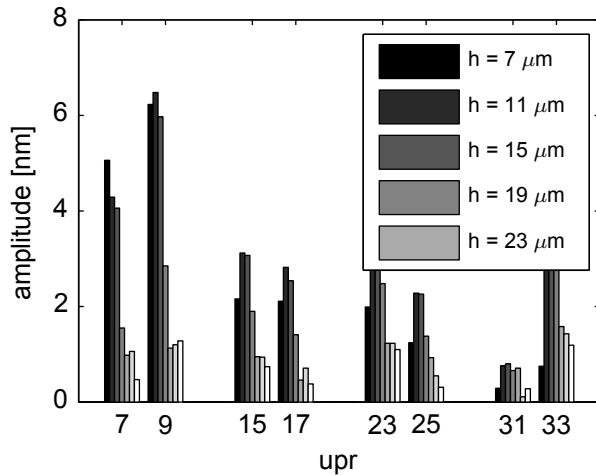


FIGURE 6. Spectrum of the radial error motion measured at different clearances (h) proving the influence of the thrust bearing on the radial error motion. Remark: these measurements include the artifact form error.

veloped that can account for the minutest details affecting the error motion and the dynamic film forces.

5-DOF MODEL

The 2-DOF film model, developed in [1], is extended to a 5-DOF film model in order to analyze the radial-, axial-, and combined radial and tilt error motion of the rotor-bearing configuration under study. The axis notation is depicted in Fig. 3. It consists of a rotor-artifact configuration with mass m that rotates around the z -axis with a speed ω of 60 rpm. The journal bearing supports the rotor radially (x - and y -direction), while the two thrust bearings provide for the axial support in the z -direction. The tilt motions (α - and β -direction) of the rotor are constrained by both the journal bearing and the tilt stiffness of the thrust bearings. The center of mass lies a distance L above the center of the journal bearing.

Equations of motion (steady-state)

Let us assume a symmetrical system such that the following set of equations can be formulated to describe the dynamic behavior of the rotor with respect to its static equilibrium position [4]:

$$k_{xx}x + k_{xy}y = f_x \quad (1)$$

$$k_{yx}x + k_{yy}y = f_y \quad (2)$$

$$(k_{zz}^{(1)} + k_{zz}^{(2)})z = f_z \quad (3)$$

$$(k_{\alpha\alpha}^{(1)} + k_{\alpha\alpha}^{(2)})\alpha + (k_{\alpha\beta}^{(1)} + k_{\alpha\beta}^{(2)})\beta + \dots \quad (4)$$

$$\dots + k_{xx}L^2\alpha + k_{xy}L^2\beta = m_\alpha$$

$$(k_{\beta\beta}^{(1)} + k_{\beta\beta}^{(2)})\beta + (k_{\beta\alpha}^{(1)} + k_{\beta\alpha}^{(2)})\alpha + \dots \quad (5)$$

$$\dots + k_{yy}L^2\beta + k_{yx}L^2\alpha = m_\beta$$

in which k_{ij} , the film stiffness, represents the film reaction in the i -direction to a displacement in the j -direction, including the phenomenon of cross-coupling. The displacement and rotation of the rotor in respectively the x -, y -, z -direction and α -, β -direction is defined accordingly. Lastly, f_x , f_y , f_z and m_α , m_β characterizes respectively the external forces and moments acting on the rotor. The indices $^{(1)}$ and $^{(2)}$ denote respectively the thrust bearing above and under the flange of the rotor.

RESULTS

The running accuracy of the rotor is analyzed using the 5-DOF film model for three different feeding systems of the rotary table, namely:

- Both journal and thrust bearings are made up of inherent restrictors
- Porous journal bearing, thrust bearings of inherent restrictors
- Porous journal and thrust bearings (full porous bearing)

The results are summarized in Fig. 7. The difference between the radial error motion of the 2-DOF film model (crosses) and the 5-DOF film model (circles) clearly indicates that the thrust bearings, and the resulting tilt error, has a detrimental effect on the running accuracy of the rotor. Moreover, the radial error motion decreases as the number of feedholes increases, as already shown in Fig. 1. A porous journal bearing reduces the radial error motion (squares) and, in the ultimate case, till sub-nanometer level for an ideal full porous bearing (triangles). Furthermore, by extrapolating the data points linearly in function of the number of feedholes, it appears that a quasi porous situation may be achieved using about 16 to 18 feedholes. The same trends can be seen for the axial error motion and the combined radial and tilt error motion. Finally, from the middle figure, one can conclude that the axial error motion is only affected by the thrust bearings.

CONCLUSIONS

In this work, the influence and the interaction are studied of the different air bearings of an axis of rotation system on the running accuracy. To this end, a coupled 5-DOF film model is developed. Firstly, one can conclude that the influence of the thrust bearings on the running accuracy cannot be ignored. Secondly, a smoother pressure profile, which can be achieved by increasing the number of feedholes, both for the journal and thrust bearings, has a favorable effect on the radial error motion of the rotor. A full porous bearing has a radial error motion of sub-nanometer level. This will be validated experimentally in a future study.

ACKNOWLEDGEMENT

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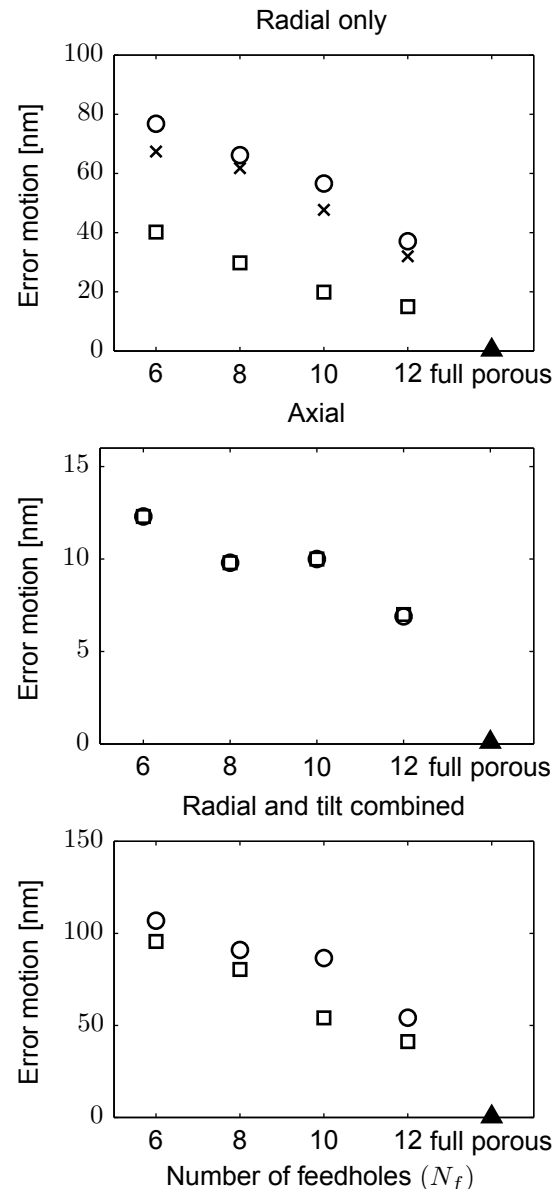


FIGURE 7. The error motion as function of the number of feedholes for different feeding systems under study.

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DESIGN OF A PRECISION FLEXURE-BASED VIBRATION MICROTOME FOR WHOLE ORGAN IMAGING

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In this paper, we present the modeling, design, and dynamic characterization of a flexure-based vibrating blade microtome that achieves submicron precision with minimized parasitic motions in multiple axes. The completed microtome is then integrated with a two-photon microscope (TPM) to form a serial two-photon tomography system for whole organ imaging. The automated system can obtain 3-D organ image at the speed of 10 GB/hr with subcellular resolution.

MICROTOME

A microtome is a sectioning instrument that enables cutting of thin slices of fresh or fixed tissue for microscopy studies or pathological analyses. Oscillating blade microtomes, or vibrating blade microtomes, are a variation of the basic microtome, and are widely recognized as superior for cutting thick sections from fixed or fresh tissue samples. Important design parameters for optimizing the cutting process include the vibration amplitude, the vibrating speed, the angle of the blade, and the feed rate (of tissue samples). However, even with proper parameter selection, conventional oscillating blade microtomes can fail to achieve the desired precision, such as precise control of tissue flatness to less than 1-2 μm and consistent section thickness. Contributing factors include: (1) imprecise oscillatory motion caused by the out-of-plane parasitic motion, (2) backlash generated in the conventional mechanical joints, and (3) friction limited frequency and magnitude for leaf spring type microtomes [1, 2]. The error motions in these systems are typically on the order of tens of microns in different axes.

DESIGN CONCEPT

We have been developing a flexure-based vibrating blade microtome device that can

address the aforementioned problems and achieve submicron cutting precision. When a vibrating blade microtome is in operation, the sample is fed to the blade in a direction, i.e. cut direction, transverse to the oscillating direction of the blade. To achieve high cut quality, the blade must oscillate in one plane, generating repeatable linear or arcuate motions. The transverse blade motion is typically generated by a motor or electromagnetic actuator coupled with an eccentric cam. A mechanical filter is required to isolate the non-transverse motions from the blade. The block diagram in Fig 1 illustrates how this can be achieved by arranging two flexures in series, where flexure 1 is stiff in the cut direction and flexure 2 is stiff in the transverse direction. The blade assembly is mounted on flexure 1.

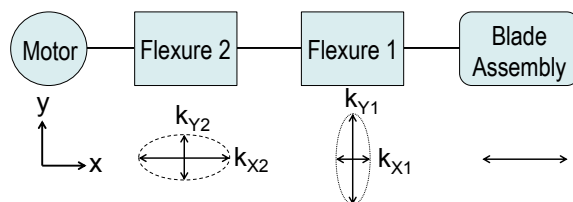


FIGURE 1. Application of two serial flexures with different stiffness ratios to isolate unwanted motions in the Y direction.

Figure 2 shows the final design of the vibrating blade microtome, where flexure 1 guides the motion of the blade and is coupled with flexure 2 that significantly reduce the unwanted mechanical coupling between the motor and flexure 1. The optimized stiffness ratio of flexure 1 and 2, i.e. k_{Y1}/k_{X1} and k_{X2}/k_{Y2} in Fig 1, are 49 and 3500 respectively. The flexures were fabricated from a 1" thick 7075 aluminum bar using wire-EDM. The 1" thickness was selected

to ensure the microtome is stiff in the out-of-plane direction during cutting processes.

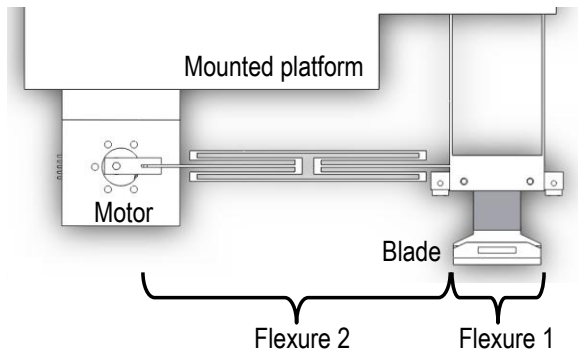


FIGURE 2. Top view of the vibrating blade microtome assembly (US patent pending).

CHARACTERIZATION

An ideal cutting operation would generate only horizontal translations of the blade, with no motion in any other axes. In practice the blade will undergo slight parasitic motions in other axes during operations due to loading from the actuators, finite stiffness of the structure and dynamic effects. As shown in Fig 3, a metrology stage was assembled around the cutting stage in order to study the motions in the axes considered most critical for cut quality. The measurement was carried out using five capacitive sensors, sampling at 2 kHz, each aligned to a 1.2 x 1.2 cm² grounded metal target which had been adhered to the moving stage.

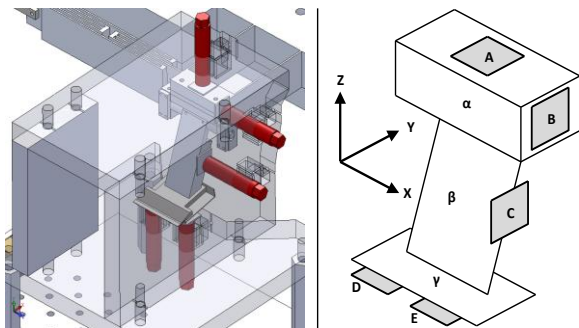


FIGURE 3. CAD model of the experimental setup, showing sensors (in red) measuring off of the vibratome stage (left). Schematic of sensor target placements as well as coordinate frame and structure labels (right).

The data were collected when operated at a vibration amplitude of 0.8 mm. The results show 2 types of parasitic motions of the blade assembly: (1) coupled Δ_z and θ_x rotation occurring on the cutting edge at 40Hz, and (2) θ_y rotation occurring with a 5° phase delay,

suggesting a rotational resonance that lies at higher frequencies than that of the θ_x rotation. These parasitic motions are illustrated in Fig. 4 A and B. The total error motion of the blade is the superposition of both of the dynamic modes described above. The net parasitic motions at the center of the blade and at the extreme tips of the blade are 0.5 μ m and -1/+2 μ m respectively. The results show the high stiffness ratios of flexure 1 and flexure 2 have effectively isolated the unwanted motions from the motor. High quality cross-sectional images can thus be obtained and benefitted from smooth microtome cuts, shown in the next section.

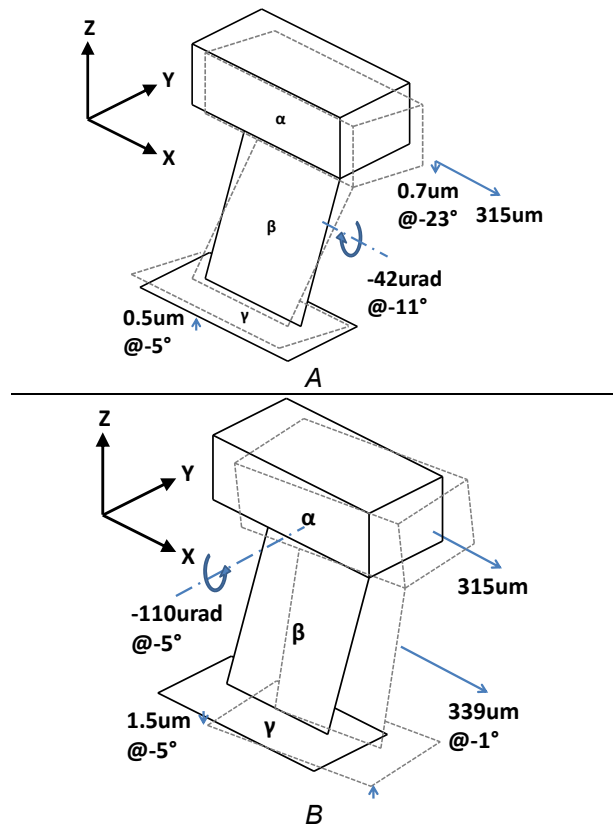


FIGURE 4. Coupled Δ_z and θ_x dynamic motion occurring on cutting stage driven at 40 Hz.

SYSTEM SETUP & IMAGING EXPERIMENTS

The completed vibrating blade microtome was integrated with a TPM system [3, 4], and an automated xyz stage to form a serial two-photon tomography system for 3-D imaging, as shown in Fig. 5.

Imaging Procedure

The imaging procedure of the system [5] is fully automated and illustrated in Fig. 6. The procedure includes: (1) sample preparation, an

animal organ such as mouse brain is first embedded in agar and then fixed in a water bath on the xyz stage under the objective lens; (2) the xyz stage moves the organ under the objective lens so that an optical section is imaged as a mosaic of fields of view; (3) the vibrating blade microtome then mechanically removes a tissue section of approximately 50-100 μ m from the top, and (4) the steps of overlapping optical and mechanical sectioning are repeated until the entire dataset is collected.

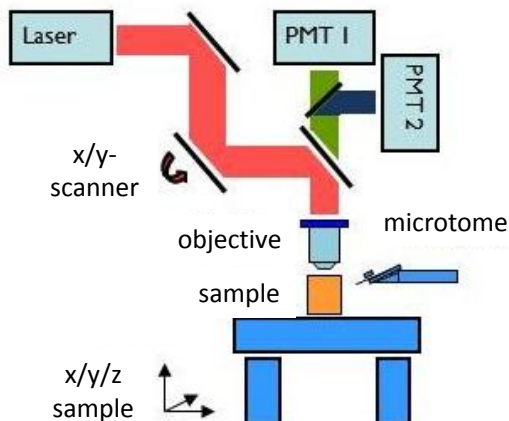


FIGURE 5. Schematics of the serial two-photon tomography system [5].

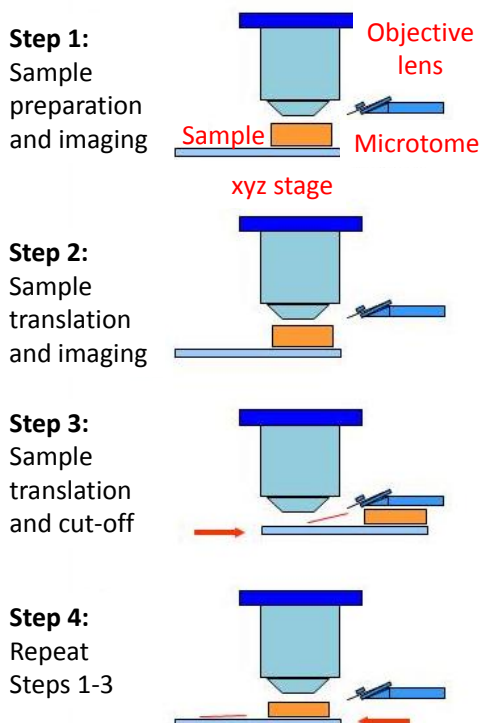


FIGURE 6. Image acquisition procedures of the serial two-photon tomography system.

Reconstructed 3-D Images of a Mouse Brain

Figure 7 – 10 present the images of a mouse brain, collected and reconstructed by the serial two-photon tomography system. Figure 7 shows the isometric view of the brain with a cut in the coronal plane. Figure 8 shows the top view of the coronal section. Figure 9 shows a zoom-in view of Fig. 8 in the upper right corner, demonstrating the subcellular resolution of the TPM system. The neurons in the brain are labeled with YFP.

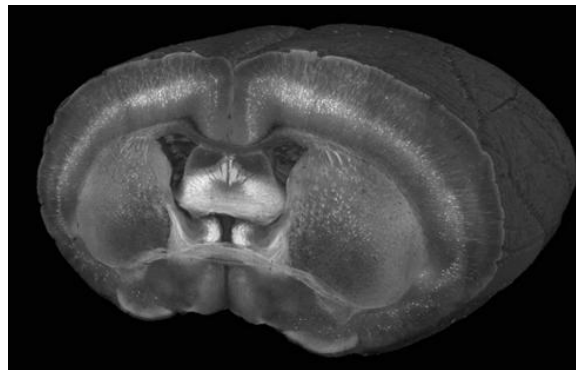


FIGURE 7. A reconstructed 3-D brain image.

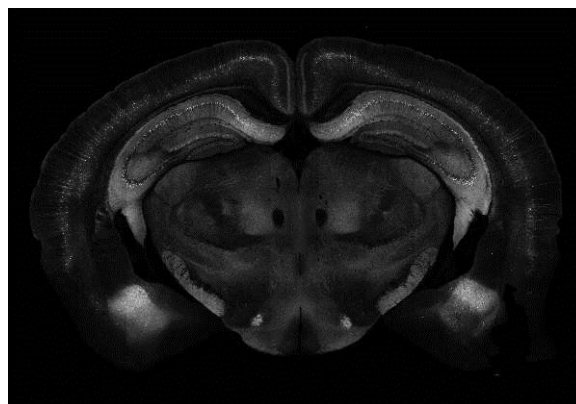


FIGURE 8. A cross-section of a C57Bl/6 mouse brain.

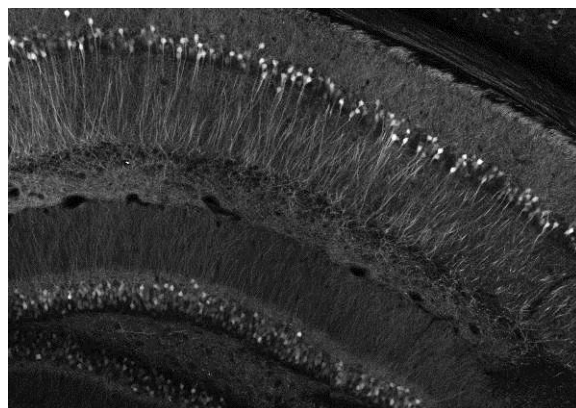


FIGURE 9. A zoom-in image of Figure 8 in the upper right corner.

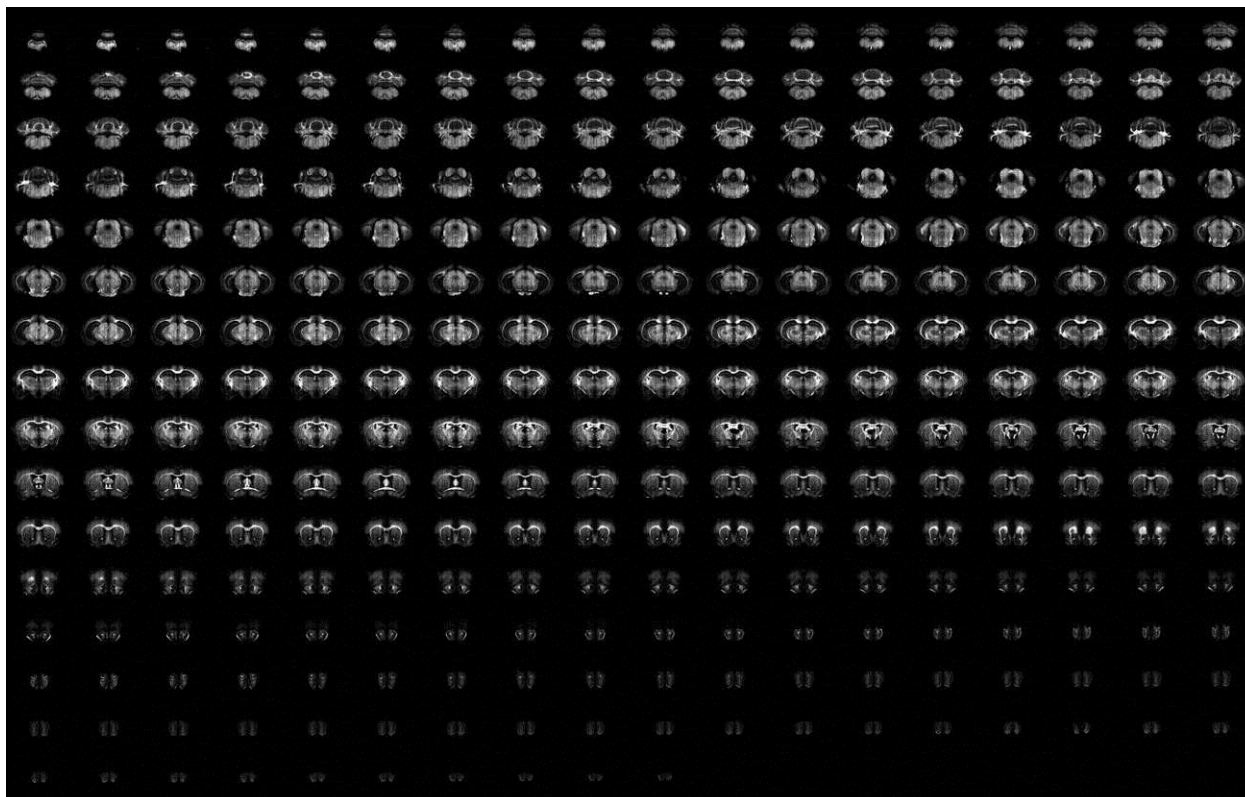


FIGURE 10. A stack of cross-sectional images from an YFP labeled single mouse brain in the coronal plane.

Figure 10 shows a series of cross-sectional images from an YFP labeled single mouse brain in the coronal plane. As images are formed by individual scanned points in space, brain slices in any other planes/orientations may be formed through post processing. These 3-D images enable studying the entire organ at subcellular resolution. In this example, the imaging of an entire brain was completed in a relatively short amount of time, i.e. 5 hours.

CONCLUSION

We have developed a flexure-based vibrating blade microtome and fully characterized its dynamic performance. The microtome shows superior precision and dynamic characteristics than existing commercial models. The improved cut quality and surface flatness significantly improve the 3-D image quality of the serial two-photon tomography system.

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UNCERTAINTY ANALYSIS OF GEOMETRIC ERROR OF SUPER-SIZE FLOOR TYPE BORING MACHINE

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ABSTRACT

Uncertainty estimation of the geometric error model of a super-size floor type boring machine is conducted through statistical analysis. Uncertainty estimation method is based on GUM and the previously developed geometric error model. Volumetric and boring error analyses are conducted to verify machining error uncertainty through the developed method.

INTRODUCTION

Many studies regarding machine tool errors and compensatory error control methods have been conducted to achieve high precision machining. Shape and joint transformation matrices have been widely applied to predict geometric errors of machine tools through kinematic chain [1-3]. However, the error prediction models have error themselves and could not perfectly compensate for the errors. These are due to uncertainty contribution of geometric error models of machine tools.

In this paper, uncertainty concept has been applied to confirm reliability of geometric error models. To estimate uncertainty contribution on the error model, stochastic input parameters are considered as probability density functions (PDF). Propagation of uncertainties of input parameters are statistically estimated through the GUM. Concept of uncertainty is proposed in the "Guide to the expression of Uncertainty in Measurement" (GUM) [4]. It is based on the law of propagation of uncertainties as shown in Fig. 1. The output quantity follows a normally scaled distribution to calculate the best estimate of input measurements and the expanded uncertainty. Several steps are required to estimate the expanded uncertainty [5-6]. Selection of input parameters in the uncertainty model and evaluation of standard uncertainty are required of the mathematical model. Standard uncertainties are propagated to the combined uncertainty through sensitivity coefficients. Expanded uncertainty is computed

by adjusting coverage factor k to the combined uncertainty. Reliability of the expanded uncertainty depends upon the coverage factor.

In this paper, uncertainty estimation from the geometric error model of a super-size floor type boring machine is studied through statistical analysis. Comparing with deterministic error models [1-3], uncertainty error model includes uncertainty sources in the measurement and the model parameters such as thermal expansion gains, thermal expansion time constants, temperature, positioning, straightness, pitch, yaw, roll and squareness errors, etc. In addition, simplification of the uncertainty error model is conducted to simplify the model from many input parameters through the selection of dominant input parameters according to the level of uncertainty contribution. Uncertainty estimation of volumetric and machining errors has been verified through the developed uncertainty error model of a super-size floor type boring machine.

MATHEMATICAL MODEL

Figs. 2 and 3 show schematic diagram of the super-size floor type boring machine under development, and its kinematic mechanism, respectively [3]. Homogeneous transformation matrices of shape and joint elements are applied for geometric error modeling of the machine tool. Volumetric error components of Eq. (1), dX , dY

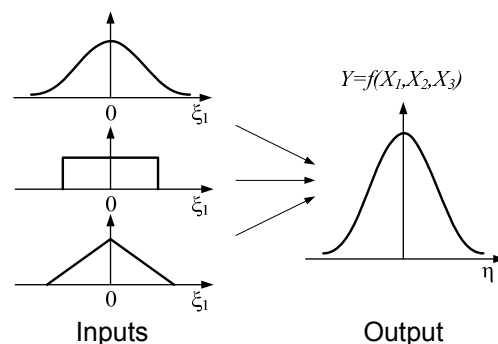


FIGURE 1. Propagation of Distributions.

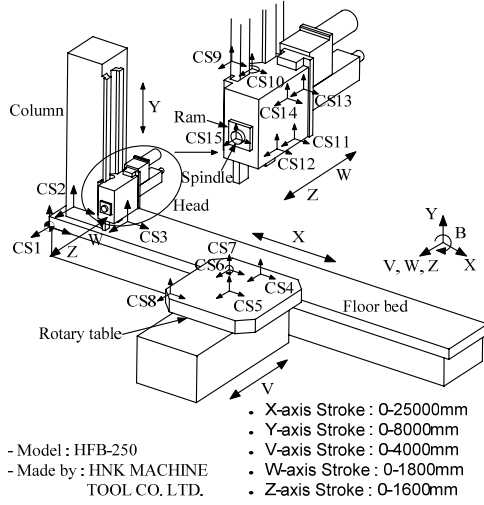


FIGURE 2. Schematic Diagram of the Super-Size Floor Type Boring Machine.

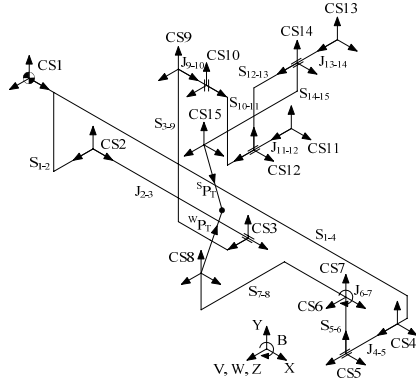


FIGURE 3. Kinematic Chain of the Super-Size Floor Type Boring Machine.

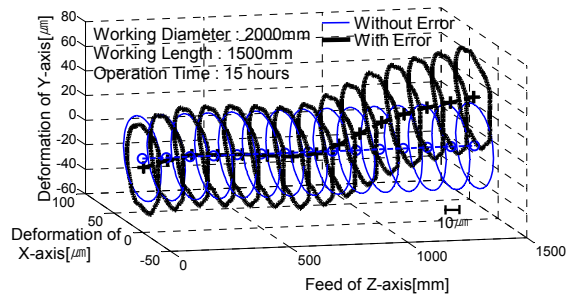


FIGURE 4. Machining Error in Boring Process.

and dZ at the tool center point $[X, Y, Z]$ are given by [3]

$$\begin{aligned} {}^wP_T &= [X+dX \ Y+dY \ Z+dZ \ 1]^T \\ &= [S_{7-8}]^{-1} [J_{6-7}]^{-1} [S_{5-6}]^{-1} [J_{4-5}]^{-1} [S_{1-4}]^{-1} \\ &\quad \times [S_{1-2}]_{actual} [J_{2-3}]_{actual} [S_{3-9}]_{actual} [J_{9-10}]_{actual} [S_{10-11}]_{actual} \\ &\quad \times [J_{11-12}]_{actual} [S_{12-13}]_{actual} [J_{13-14}]_{actual} [S_{14-15}]_{actual} {}^SP_T \end{aligned} \quad (1)$$

$$[S_{1-2}]_{actual} = \begin{bmatrix} 1 & -\varepsilon_{21}^s(T) & \varepsilon_{21}^s(T) & a_1 + \Delta a_1(T) \\ \varepsilon_{21}^s(T) & 1 & -\varepsilon_{21}^s(T) & b_1 + \Delta b_1(T) \\ -\varepsilon_{21}^s(T) & \varepsilon_{21}^s(T) & 1 & c_1 + \Delta c_1(T) \\ 0 & 0 & 0 & 1 \end{bmatrix} \quad (2)$$

$$[J_{13-14}]_{actual} = \begin{bmatrix} 1 & -\varepsilon_{16}(z, T) & \varepsilon_{16}(z, T) & \delta_1(z, T) \\ \varepsilon_{16}(z, T) & 1 & -\varepsilon_{16}(z, T) & \delta_1(z, T) - S_{16} \cdot z \\ -\varepsilon_{16}(z, T) & \varepsilon_{16}(z, T) & 1 & z + \delta_1(z, T) + P_1(T) \cdot z \\ 0 & 0 & 0 & 1 \end{bmatrix} \quad (3)$$

In Eq. (1), error vector consists of 8 shape and 6 joint elements. Eqs. (2) and (3) show ones of the shape and joint transform elements, respectively. The input parameters, rotational errors of shape ε_{ij}^s , rotational errors of joint ε_{ij} , translational errors due to thermal deformation of shape elements $\Delta a_i, b_i, c_i$, linear thermal expansion of a feed axis P_i , and positioning or straightness errors of feed axes δ_i are function of temperature. Fig. 4 shows machining errors generated in the boring process. However, there is uncertainty due to measurement and environmental effects as well as modeling error. It is impossible to propose uncertainty contribution and reliability of the geometric error given by Eq. (1).

TABLE 1. Input Parameters of Uncertainty Model.

Thermal Expansion Gain
$u_{D,a1}, u_{D,b1}, u_{D,c1}, u_{D,a2}, u_{D,b2}, u_{D,c2}$ $u_{D,a3}, u_{D,b3}, u_{D,c3}, u_{D,b4}, u_{D,a5}, u_{D,b5}, u_{D,c5}$ $u_{D,a6}, u_{D,b6}, u_{D,c6}, u_{D,a7}, u_{D,b7}, u_{D,c7}$ $u_{D,a8}, u_{D,b8}, u_{D,c8}, u_{D,x}, u_{D,y}, u_{D,v}, u_{D,w}, u_{D,z}$
Thermal Expansion Time Constant
$u_{\tau,1}, u_{\tau,2}, u_{\tau,3}, u_{\tau,4}, u_{\tau,5}, u_{\tau,6}, u_{\tau,7}, u_{\tau,8}$ $u_{\tau,x}, u_{\tau,y}, u_{\tau,v}, u_{\tau,w}, u_{\tau,z}$
Temperature, Positioning, Straightness
$u_{Temperature}, u_{\delta,x}, u_{\delta,y}, u_{\delta,v}, u_{\delta,w}, u_{\delta,z}, u_{\delta,x,y}, u_{\delta,x,v}$ $u_{\delta,x,w}, u_{\delta,x,z}, u_{\delta,y,x}, u_{\delta,y,v}, u_{\delta,y,w}, u_{\delta,y,z}, u_{\delta,z,x}, u_{\delta,z,y}$
Angular Error, Squareness Error
$u_{\varepsilon,a1}, u_{\varepsilon,b1}, u_{\varepsilon,c1}, u_{\varepsilon,a2}, u_{\varepsilon,b2}, u_{\varepsilon,c2}, u_{\varepsilon,a3}, u_{\varepsilon,b3}$ $u_{\varepsilon,c3}, u_{\varepsilon,a4}, u_{\varepsilon,b4}, u_{\varepsilon,c4}, u_{\varepsilon,a5}, u_{\varepsilon,b5}, u_{\varepsilon,c5}, u_{\varepsilon,a6}$ $u_{\varepsilon,b6}, u_{\varepsilon,c6}, u_{\varepsilon,a7}, u_{\varepsilon,b7}, u_{\varepsilon,c7}, u_{\varepsilon,a8}, u_{\varepsilon,b8}, u_{\varepsilon,c8}$ $u_{\varepsilon,ax}, u_{\varepsilon,bx}, u_{\varepsilon,cx}, u_{\varepsilon,ay}, u_{\varepsilon,by}, u_{\varepsilon,cy}, u_{\varepsilon,av}, u_{\varepsilon,bv}$ $u_{\varepsilon,cv}, u_{\varepsilon,aw}, u_{\varepsilon,bw}, u_{\varepsilon,cw}, u_{\varepsilon,az}, u_{\varepsilon,bz}, u_{\varepsilon,cz}, u_{S,xy}$ $u_{S,vx}, u_{S,vy}, u_{S,zy}$

INPUT PARAMETERS OF A MODEL

To investigate uncertainty contributions on the modeled volumetric error given by Eq. (1), uncertainty sources are to be identified first. They are to be included in the volumetric error model. There are 99 uncertainty sources such as 8 thermal expansion gains of shape elements, 5 thermal expansion gains of joint elements, 8 thermal expansion time constants of shape elements, 5 thermal time constants of joint elements, a temperature, 5 positioning errors, 10 straightness errors, 24 parameters on pitch, yaw roll errors of shape elements, 15 parameters on pitch, yaw roll errors of joint elements, and 4 parameters on squareness errors. Table 1 shows the input parameters of the uncertainty model.

STANDARD UNCERTAINTY EVALUATION

In order to estimate uncertainty contribution through measurement, standard uncertainty of input parameters on the uncertainty model is computed through the A-type uncertainty evaluation method. However, as the floor type boring machine is fabricating in the factory, it is impossible to measure A-type uncertainty components. Instead of A-type uncertainty, B-type uncertainty evaluation is applied in this paper. It is assumed that input parameters of the uncertainty model are uniformly distributed with range $\pm R/2$ to define the standard uncertainty as shown in Fig. 5. Distribution range R of the thermal expansion gain is selected with $\pm 10\%$ for model identification error. R of temperature error is selected with $\pm 1.6^\circ\text{C}$ to consider temperature gradient according to the machine height difference. Standard uncertainty of positioning, straightness, pitch, yaw, roll and squareness errors depends on machine tool metrology. They are selected with 10 times of laser measurement resolution such as $0.01\mu\text{m}$ for positioning, $1\mu\text{m}$ for straightness, $1\mu\text{m/m}$ for pitch, yaw and roll, and $0.1\mu\text{m/m}$ for squareness errors.

COMBINED UNCERTAINTY

Eqs. (4) and (5) show the propagation of the standard uncertainty and sensitivity coefficient of each standard uncertainty factor, respectively. x is an input parameter of the error model and y is the output. 99 uncertainty factors are combined together with sensitivity coefficients to make combined uncertainties in x, y and z directions. Eq. (6) shows the combined uncertainty in x direction.

PARAMETER SIMPLIFICATION

As the sensitivity coefficients have different contributions to the combined uncertainty, the uncertainty model is able to be simplified with dominant sensitivity parameters. Fig. 6 shows 99 uncertainty contributions according to input parameters of the volumetric error model. Model simplification is conducted by filtering the input parameters according to the contribution on the combined uncertainty. 33 input sensitivities are selected from 99 input parameters, and finally 15, 17 and 18 input parameters are selected to estimate the combined uncertainties in x, y and z directions, respectively. Eq. (7) shows the simplified combined uncertainty in x direction. Similar results are obtained in y and z directions.

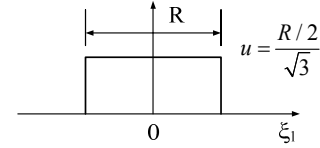


FIGURE 5. Uniformly Distributed PDF.

$$u_c^2(y) = \sum_{i=1}^N c_i^2 u^2(x_i) \quad (4)$$

$$c_i = \left[\frac{\partial f(\xi_1, \dots, \xi_N)}{\partial \xi_i} \right] \quad (5)$$

$$\begin{aligned} u_{c,x}^2 = & c_{X,D,a1}^2 u_{D,a1}^2 + c_{X,D,a2}^2 u_{D,a2}^2 + c_{X,D,a3}^2 u_{D,a3}^2 \\ & + c_{X,D,a5}^2 u_{D,a5}^2 + c_{X,D,a6}^2 u_{D,a6}^2 + c_{X,D,a7}^2 u_{D,a7}^2 \\ & + c_{X,D,a8}^2 u_{D,a8}^2 + c_{X,D,x}^2 u_{D,x}^2 + c_{X,\tau,1}^2 u_{\tau,1}^2 \\ & + c_{X,\tau,2}^2 u_{\tau,2}^2 + c_{X,\tau,3}^2 u_{\tau,3}^2 + c_{X,\tau,5}^2 u_{\tau,5}^2 \\ & + c_{X,\tau,6}^2 u_{\tau,6}^2 + c_{X,\tau,7}^2 u_{\tau,7}^2 + c_{X,\tau,8}^2 u_{\tau,8}^2 \\ & + c_{X,\tau,x}^2 u_{\tau,x}^2 + c_{X,Temperature}^2 u_{Temperature}^2 \\ & + c_{X,\delta,x}^2 u_{\delta,x}^2 + c_{X,\delta,x,y}^2 u_{\delta,x,y}^2 + c_{X,\delta,x,v}^2 u_{\delta,x,v}^2 \\ & + c_{X,\delta,x,w}^2 u_{\delta,x,w}^2 + c_{X,\delta,x,z}^2 u_{\delta,x,z}^2 + c_{X,e,b1}^2 u_{e,b1}^2 \\ & + c_{X,e,c1}^2 u_{e,c1}^2 + c_{X,e,b2}^2 u_{e,b2}^2 + c_{X,e,c2}^2 u_{e,c2}^2 \\ & + c_{X,e,b3}^2 u_{e,b3}^2 + c_{X,e,c3}^2 u_{e,c3}^2 + c_{X,e,b4}^2 u_{e,b4}^2 \\ & + c_{X,e,c4}^2 u_{e,c4}^2 + c_{X,e,b5}^2 u_{e,b5}^2 + c_{X,e,c5}^2 u_{e,c5}^2 \\ & + c_{X,e,b6}^2 u_{e,b6}^2 + c_{X,e,c6}^2 u_{e,c6}^2 + c_{X,e,b7}^2 u_{e,b7}^2 \\ & + c_{X,e,c7}^2 u_{e,c7}^2 + c_{X,e,b8}^2 u_{e,b8}^2 + c_{X,e,bx}^2 u_{e,bx}^2 \\ & + c_{X,e,cx}^2 u_{e,cx}^2 + c_{X,e,by}^2 u_{e,by}^2 + c_{X,e,bv}^2 u_{e,bv}^2 \\ & + c_{X,e,cv}^2 u_{e,cv}^2 + c_{X,e,bw}^2 u_{e,bw}^2 + c_{X,e,cw}^2 u_{e,cw}^2 \\ & + c_{X,e,bz}^2 u_{e,bz}^2 + c_{X,e,cz}^2 u_{e,cz}^2 + c_{X,S,xy}^2 u_{S,xy}^2 \\ & + c_{X,S,vx}^2 u_{S,vx}^2 \end{aligned} \quad (6)$$

$$\begin{aligned} u_{c,x}^2 = & c_{X,D,a2}^2 u_{D,a2}^2 + c_{X,D,x}^2 u_{D,x}^2 + c_{X,\tau,1}^2 u_{\tau,1}^2 \\ & + c_{X,\tau,2}^2 u_{\tau,2}^2 + c_{X,\tau,3}^2 u_{\tau,3}^2 + c_{X,\tau,5}^2 u_{\tau,5}^2 \\ & + c_{X,\tau,6}^2 u_{\tau,6}^2 + c_{X,\tau,7}^2 u_{\tau,7}^2 + c_{X,\tau,8}^2 u_{\tau,8}^2 \\ & + c_{X,\tau,x}^2 u_{\tau,x}^2 + c_{X,\delta,x,y}^2 u_{\delta,x,y}^2 + c_{X,\delta,x,v}^2 u_{\delta,x,v}^2 \\ & + c_{X,\delta,x,w}^2 u_{\delta,x,w}^2 + c_{X,\delta,x,z}^2 u_{\delta,x,z}^2 + c_{Sim}^2 u_{Sim}^2 \end{aligned} \quad (7)$$

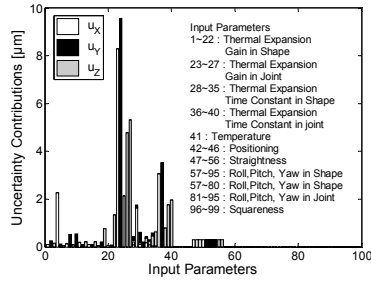


FIGURE 6. Uncertainty Contributions.

EXPANDED UNCERTAINTY

Expanded uncertainty with 95.45% of reliability is calculated from the combined uncertainty in x, y and z directions with coverage factor $k=2$. Eq. (8) shows the relationship between the expanded and combined uncertainties. Fig. 7 shows the expanded uncertainty on the volumetric error of the super-size floor type boring machine. Figure 8 shows the machining uncertainty occurred in the boring process. Comparing Fig. 8 with Fig. 4, it is confirmed that machining uncertainty generated in the boring process is estimated well through the proposed simulation. Fig. 9 shows the expanded uncertainty of the positioning error in x direction.

$$U_X = k \cdot u_{c,X} \quad U_Y = k \cdot u_{c,Y} \quad U_Z = k \cdot u_{c,Z} \quad (8)$$

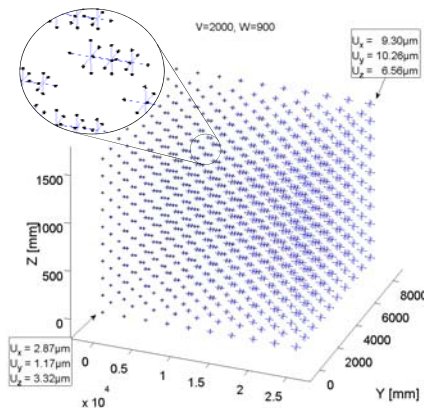


FIGURE 7. Uncertainty of Volumetric Errors.

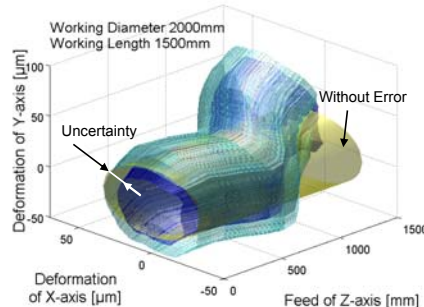


FIGURE 8. Boring Errors with Uncertainty.

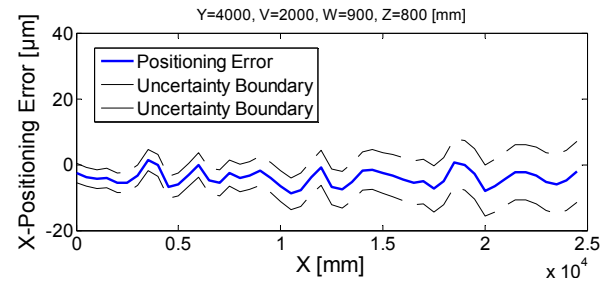


FIGURE 9. Positioning Error with Uncertainty.

CONCLUSIONS

Uncertainty modeling of machining error generated from a super-size floor type boring machine is proposed through GUM concept. Through the uncertainty contribution analysis according to sensitivity coefficient of each standard uncertainty factor, a simplified uncertainty model of the geometric error has been obtained. Uncertainty of volumetric, boring and positioning errors is estimated based on B-type uncertainty of input parameters. It is confirmed that the proposed uncertainty model has 95.45% reliability.

ACKNOWLEDGEMENT

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DESIGN & SIMULATION OF A PRECISION ROTATING VALVE SYSTEM FOR NEUTRON IMAGING

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INTRODUCTION

The development of neutron imaging at Lawrence Livermore National Laboratory (LLNL) has led to the development of various neutron generating devices. From rotating apertures, to pulse valve gas systems, to hard targets, all have worked by placing an interaction material in front of a deuteron beam to generate neutrons. The next generation of neutron generating devices will utilize a precision rotating gas valve system that will place a deuterium gas pulse in front of deuteron beam at a rate of 120Hz. As with the other neutron generating devices, it will be part of a larger system that will include a linear accelerator that creates the deuteron beam, a scintillator detector and high resolution CCD cameras.

SYSTEM DESIGN

The rotating valve design consists of a rotating valve encased in a pressure vessel as shown below in Figure 1. The valve is feed by intake and exhaust ports that carry deuterium and argon gases.

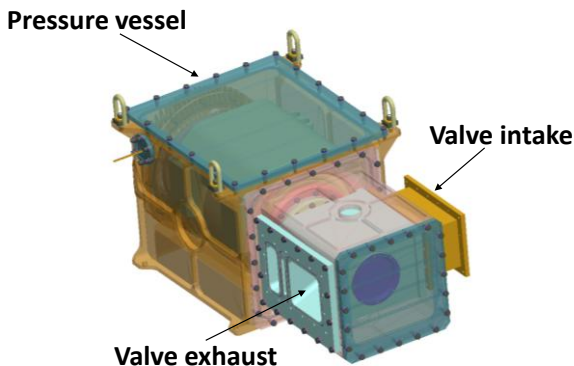


FIGURE 1. Rotary Valve in Pressure Vessel.

A cutaway of the front end of the valve is shown in Figure 2 with the rotary valve lined up with the intake and exhaust ports. This is the position in which the beam is fired through deuterium and argon chambers. The valve head rotates within the valve housing with a targeted radial gap of

50 microns. This will limit leakage into the pressure vessel.

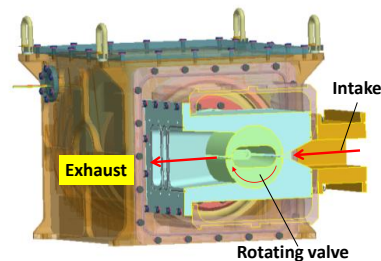


FIGURE 2. Cutaway of valve front end.

A complete cutaway along the beam line is shown below in Figure 3.

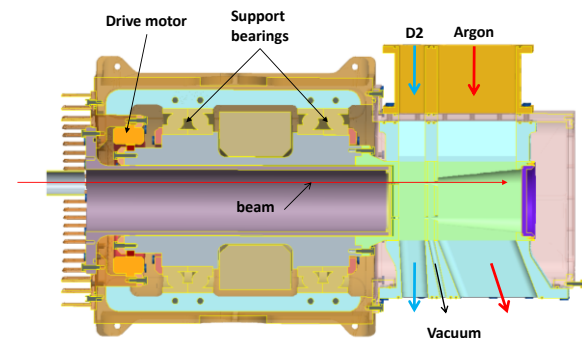


FIGURE 3. Cutaway showing chambers.

Every half revolution the beam will fire through a vacuum beam line tube that runs between the support bearings and is ultimately connected to the deuteron linear accelerator. As the valve ports line up with the chambers, a shutter in the beam tube is uncovered to allow the beam to fire through from the beam tube and onto the valve chambers for 150 microseconds. The first chamber the beam enters is filled with deuterium (D2) at a pressure of 5 bar and the beam interaction with this gas leads to the creation of neutrons. The beam next fires through to the intermediate chamber, which is held at a low pressure so as to reduce the mixing of D2 and

argon in their respective chambers. The beam finally enters the beam stop chamber, where the beam interacts with argon at a pressure of 5 bar and causes the beam energy to be dissipated into the moving argon gas.

SYSTEM MODELING

Modeling of the valve was done in two ways: first using a lumped parameter model and later using Ansys Fluent, which is a commercial Computational Fluid Dynamic (CFD) program with a moving mesh.

For the lumped modeling the valve was modeled as a rotating slot with the gas input through a venturi at the top as shown in Figure 4. The slot width is 6mm and as the slot lines up with the intake port, the gas flows through the chamber and out through the exhaust port.

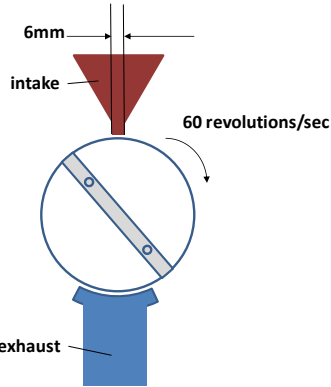


FIGURE 4. Simplified model of valve.

The rotating slot was modeled as a control volume with isentropic nozzles at either end for intake and exhaust as shown in Figure 5. The mass flow and enthalpy through the nozzles is described in equation 1 and 2 below [1]. The nozzle flow area, $A(t)$ is the time varying area of the intake or exhaust ports. The fluid flow was assumed to be inviscid for simplicity.

$$\dot{m} = A(t) \frac{P_u}{\sqrt{T_u}} \sqrt{\frac{2\gamma}{R(\gamma-1)}} \sqrt{Pr^{2/\gamma} - Pr^{(\gamma+1)/\gamma}} \quad (1)$$

$$\dot{h} = C_p T_u A(t) \frac{P_u}{\sqrt{T_u}} \sqrt{\frac{2\gamma}{R(\gamma-1)}} \sqrt{Pr^{2/\gamma} - Pr^{(\gamma+1)/\gamma}} \quad (2)$$

Where:

$\dot{m}$ = mass flow into or out of control volume

$\dot{h}$ = enthalpy flow into or out of control volume

$A(t)$ = flow area

P_u = upstream pressure

P_d = downstream pressure

C_p = specific heat constant pressure

C_v = specific heat constant volume

$$Pr = \frac{P_d}{P_u} \quad \gamma = \frac{C_p}{C_v} \quad R = C_p - C_v$$

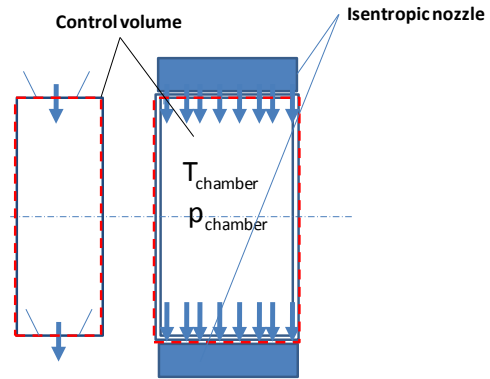


FIGURE 5. Side view of control volume.

The energy in the control volume, (cv) is calculated as changes in temperature due to the beam power in the control volume and the flow of the gas enthalpies in and out as shown below in equation (3):

$$\frac{dT_{cv}}{dt} = \frac{q + [\dot{h}_{in}(T_{cv}) - \dot{h}_{out}(T_{cv})]}{(m_{in} - m_{out})C_v} \quad (3)$$

Where:

q = beam power into control volume

T_{cv} = temperature of control volume

The beam power represents the heat output of the reaction between the deuteron beam and the gas. The pressure in the control volume is then calculated from the ideal gas law applied to the control volume as shown below in equation (4):

$$P_{cv} = \frac{m_{cv} R T_{cv}}{V_{cv}} \quad (4)$$

The complete dynamic model represented by equations 1 through 4 was simulated using helium in lieu of deuterium gas. Helium has

similar density to deuterium but different heat capacities. For the initial experiments helium will be used in-lieu of deuterium because it's not an explosive gas. Once the rotary valve system has been shown to work with helium, it would be placed in front of a linear accelerator with deuterium and argon gas sources.

The simulated pressure pulse inside the control volume is shown in Figure 6, with a 5 atmosphere helium gas source. Also shown is the rotating valve intake area as a function of time, thus showing the delay between the maximum valve opening and the peak pressure pulse.

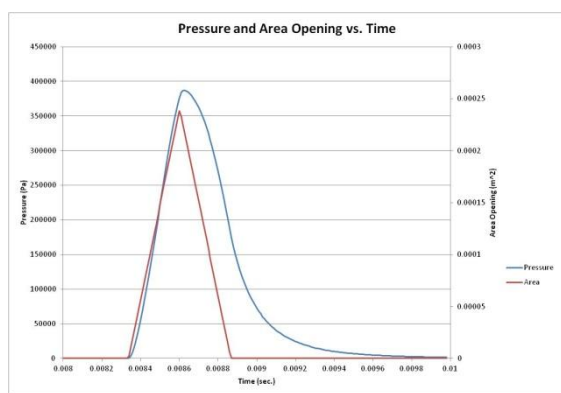


FIGURE 6. Pressure pulse simulation.

The rotary valve was also modeled using Ansys's Fluent CFD software with a moving mesh to simulate the rotation of the slot past the intake and exhaust ports as shown in Figure 7.

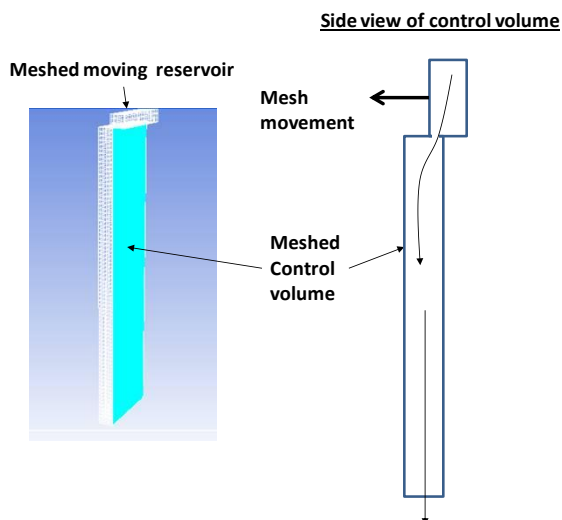


FIGURE 7. Ansys moving mesh model

The CFD model was initially run with helium under inviscid flow as way to compare it with the analytical lumped parameter model. Both models compared well with each other as shown below in Figure 8. When the more realistic condition of viscous flow was implemented deviations were seen as shown in the last 2 columns. With viscous flow comes a boundary layer that narrows the flow path through the slot, increases pressure and velocity, which leads to further expansion of the gas and cooling of the gas as it exits the chamber. These results are what are expected when the rotary valve is operated with helium.

Boundary Condition: 500 kPa inlet pressure, 30kPa outlet pressure, no power

	Analytical	CFD Inviscid Flow	CFD	% Difference
Upper Pressure	388,000 Pa peak	369,000 Pa	431,000 Pa Beam line	11%
Upper Temperature	332 K peak	362 K	257K beam line	-22%
Density at beam	0.48 Kg/m ³ Inferred From NIST table	0.49 Kg/m ³	0.58 Kg/m ³	~21 %
Upper velocity beam	*570 m/s	525 m/s	754 m/s	32%

*inviscid flow

FIGURE 8. Table comparing different simulation results for helium gas at 5 atmospheres

The ultimate goal of the rotary valve is to place helium (deuterium for actual imaging) in front of the beam with sufficient density so that the beam interaction will produce neutrons. The minimum density required for that is 0.48 Kg/m³. The CFD simulation of density at top dead center is shown below in Figure 9.

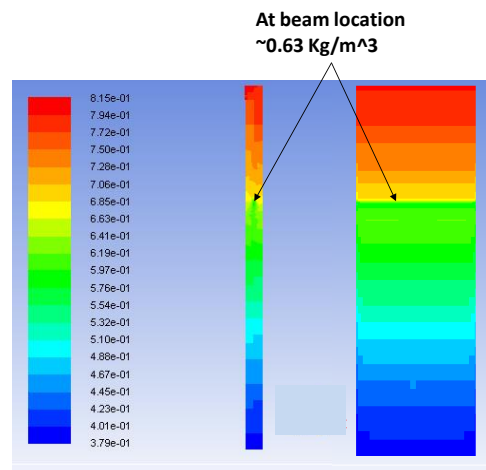


FIGURE 9. Helium density in control volume.

The predicted density is 0.63 Kg/m³, which is more than sufficient to generate neutrons.

LEAKAGE & ERROR BUDGET

The above simulations do not account for the leakage between the rotating rotary valve head and the valve head body as shown in Figure 10. The leakage represents an additional amount of gas that will need to be added to the intake flow into the rotary valve.

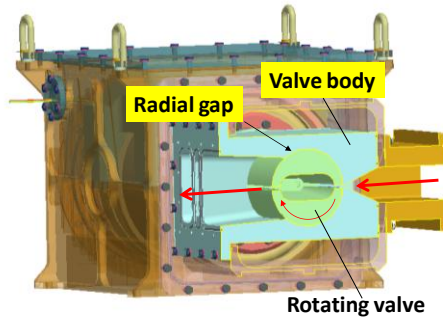


FIGURE 10. Leakage between rotating valve and valve body.

This leakage has been modeled using CFD and approximated using a simple isentropic nozzle model. These simulations track well together and with the area of the leakage as shown below in Figure 11 for a range of radial gaps.

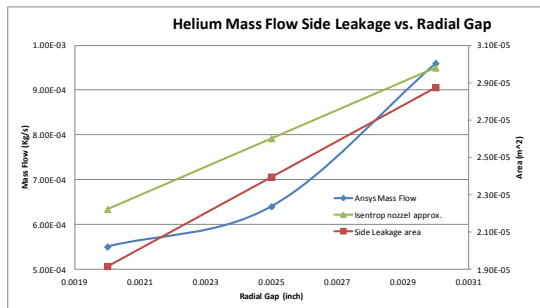


FIGURE 11. Simulations of leakage out the side of the valve as a function of radial gap.

Leakages from all sources can easily double the total mass flow needed to achieve the desired gas densities in the slot. This will increase the cost of the gas handling system. Reducing the leakage requires the radial gap to be as small as possible given fabrication constraints.

An error budget was created to estimate the possible radial gaps that could be achieved using precision machining methods as shown in Figure 12. The sources of potential error include

bearing run out, differences in bearing support diameters, rotary valve location on the main shaft, rotary valve run out, main shaft bearing surface misalignment and the tilting of the rotor due to the perpendicularity tolerance between the rotor mounting flange and the axis of rotation. In addition to the fabrication errors, the change in the radial gap due to the thermal expansion of the rotary valve was estimated using a Finite element model of the rotary valve. The boundary conditions for the model included the cooling effect of the deuterium and argon gases. The heat source was the heat generated by the argon as it stops the beam, which is estimated to be 109kW for 150 microseconds.

Error Source	Radial Direction (um)	type
Brg. run out	19.53	p-v
Brg. O.D. support	32.54	p-v
Rotor valve run out	6.35	p-v
Shaft mis-alignment	19.10	p-v
Tilt at rotor	42.95	p-v
Thermal expansion	0.26	p-v
Sum	120.73	p-v
RMS of p-v	17.56	p-v
Total (Sum+RMS)/2	69.15	p-v

FIGURE 12. Error budget for valve's radial gap.

The total error estimated was 69.15 microns, which was calculated by averaging the Sum plus RMS errors [2]. This exceeds the targeted radial gap of 50 microns. To achieve this gap, we plan to measure run out, misalignment and tilt errors of the individual components and compensate by altering tolerances and nominal dimensions as necessary. We also may have to preferentially assemble to minimize tilt errors.

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MECHANISM OF DRIVING FORCE GENERATION IN TWO-DIMENSIONAL NON-CONTACT TRANSFER METHOD BY NEAR-FIELD ULTRASONIC LEVITATION

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INTRODUCTION

Noncontact transfer devices are demanded when silicon wafers or glass plates are carried in clean environment. Because contact of such products with a conveyor belt damages and contaminates their surface, non-contact transfer devices are preferable. Many transfer devices using ultrasonic, electromagnetic or electrostatic levitation have been proposed. The ultrasonic levitation using the squeeze effect can be applied to thin plates [1]. A travelling wave was added to the levitation to transfer the plates [2][3]. A device combining two sets of a 1-directional device carries a plate in two directions [4]. The authors also have proposed a two-dimensional transfer method using ultrasonic levitation [5]. Because it utilizes repulsive force, gap sensors to measure the levitation distance are not needed. Electromagnetic properties do not affect the levitation and transfer performance.

In this paper, a driving force in a transfer device with two degrees of freedom using ultrasonic levitation is analyzed by measuring the vibration and pressure.

EXPERIMENTAL SETUP

Figure 1 shows a structure of the two-dimensional transfer device [5]. It consists of a square vibration plate and four bolted Langevin

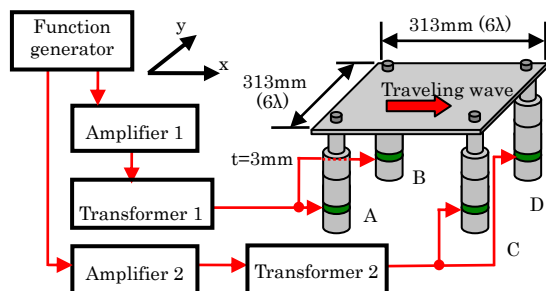


Figure 1. Two-dimensional transfer device.

transducers (BLTs) with step horns. The resonance frequency of the vibration plate was designed by the modal analysis of the finite element method. The vibration plate made of aluminum measures 313 mm (6 wavelengths at 20 kHz) in length and width, and 3 mm in thickness. The BLTs were fixed at loops of the vibration plate near its corners. In transfer experiments, two adjacent BLTs were electrically connected in parallel and driven synchronously. A step-up transformer with a turn ratio of 1:4 was cascaded to each set of the BLTs for the impedance matching to the amplifier. A two-channel function generator was used to adjust the phase between two channels. A carried plate measures 120 mm in length and width (2.3 wavelengths) and 2 mm in thickness, and weighs 81 g.

TRANSFER PERFORMANCE

When the BLTs were connected in parallel as the sets of A-B and C-D as shown in Figure 1, the plate was carried in the x-direction. When the plate was carried in the y-direction, the BLTs were connected as the sets of A-C and B-D.

Figure 2 shows the relationship between the transfer velocity and applied voltage. The voltage was changed from 100 to 1300 Vp-p by

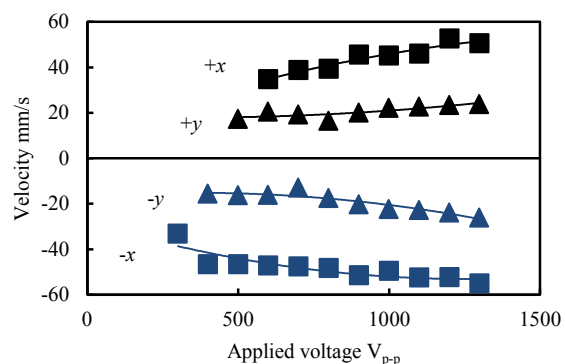


Figure 2. Relationship between transfer velocity and applied voltage.

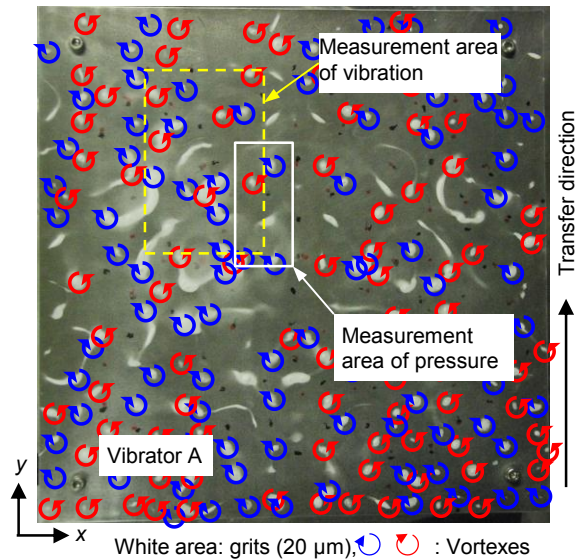


Figure 3. Vibration plate with spread grits.

a step of 100 Vp-p. The phase difference was set to ± 90 degrees. The driving frequencies were 19.90 and 19.92 kHz in the x- and y-directions, respectively. The velocity of the carried plate was measured by an image tracker after taking on video.

The carried plate was moved in desired directions over 300 Vp-p. The velocity was slightly increased with an increase of the applied voltage. The carried plate was levitated over an applied voltage of 300 Vp-p and the levitation distance was gradually increased. The threshold voltage of the transfer agreed with that of the levitation. The carried plate was actually levitated 22 μm at 500 V by measuring with an eddy current displacement sensor.

The stability of the transfer was sensitive to the driving frequency. When the driving frequency was off over 20 Hz, the motion of the carried plate was oscillatory around the center of the vibration plate. A little asymmetry of the structure or clamping torque caused the difference of the resonance frequencies in the x- and y-directions. The carried plate moved toward a corner across the center of the vibration plate.

GENERATED PRESSURE

Motion of Small Particles

Alumina grits with an average diameter of 20 μm were spread on the vibration plate to visualize a vibration mode. Figure 3 shows a result under

the driving conditions of 1300 Vp-p as voltage, 19.88 kHz as frequency and 90 deg. as phase difference, which are the conditions for the y-direction motion. The grits appearing as white areas clustered at nodes where vortices were observed. No overall stream was observed on the whole area. The vortices were condensed near the vibrators. The directions of adjacent vortices were symmetrical. Consequently, the rowed vortices might generate the driving force. When the carried plate swerved from the center line once, it moved toward a vibrator.

Displacement of Vibration Peaks

The vibration mode of the vibration plate was measured with a laser Doppler velocimeter. The measured area is shown in Figure 3 with the dashed rectangle. Figure 4 shows a transition of vibration peaks. The vibration was measured by a 2-mm pitch on the basis of the applied voltage to the vibrator A. Then the distribution of the vibration at intervals of 1/8 cycles was displayed synchronously because the vibration was cyclic. The arrows shown in the 0/8 cycle point the directions of the peak motions. The peaks revolved about the nodes. The grits moved in the reverse direction to the vibration vortices, similar to ultrasonic motors.

Although the peaks moved toward the goal side on the center line, ones mainly moved toward the left side or revolved about the nodes on the other area. The wavelength was 46 mm, which was 88% of the designed traveling wave of bending vibration.

Displacement of Pressure Peaks

The pressure was also measured by a 2-mm pitch with a strain gauge type pressure sensor with a measurement range of 100 kPa and a resonance frequency of 22 kHz. The sensor was covered with a plate with a 1.5-mm hole. The pressure signal was obtained through a low pass filter with a cut-off frequency of 40 kHz. Because the driving frequency was near the resonance frequency of the pressure sensor, the phase was shifted and the pressure amplitude might be increased. However, the trend allowed qualitative discussions. The cover plate embedding the pressure sensor was hold with pins and its in-plane looseness was 0.5 mm. The measured area is shown as the solid rectangle in Figure 3.

Figure 5 shows a change of pressure distribution. Pressure peaks cyclically moved to carrying

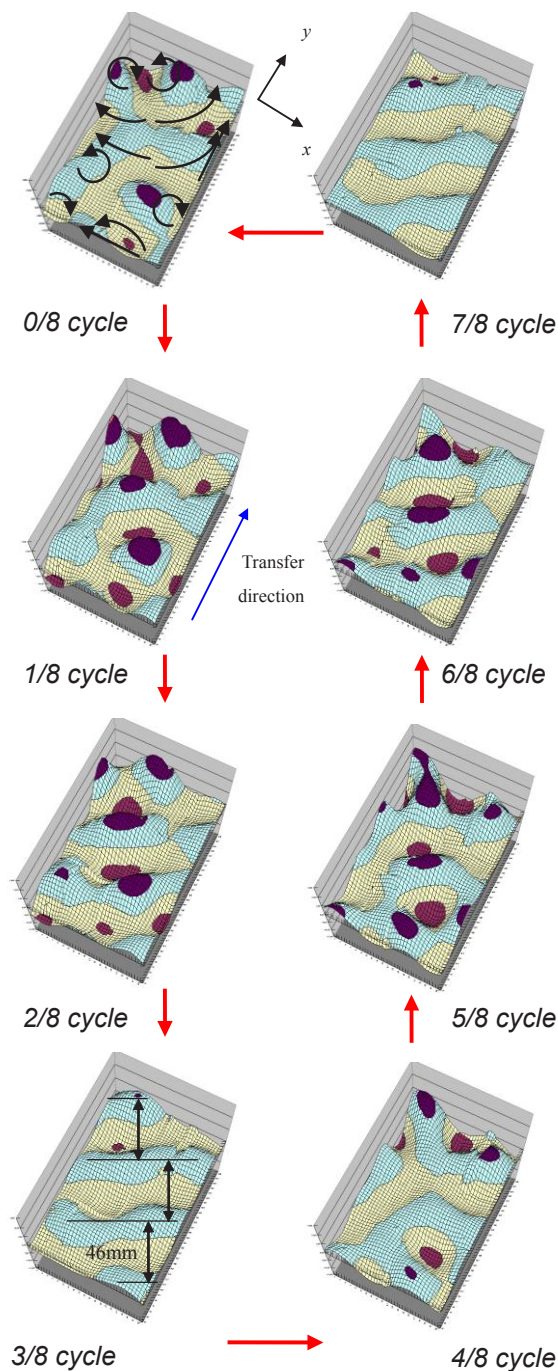


Figure 4. Transition of vibration peaks. (Measurement range: 80×120 mm; vertical scale is ±15 μm.)

direction. The arrows shown in 0/8 cycle point the moving direction of pressure peaks. The wavelength of the pressure was 47 mm, which was 90% of the designed wavelength. These coincided with the direction and wavelength of the traveling wave of the plate vibration. The

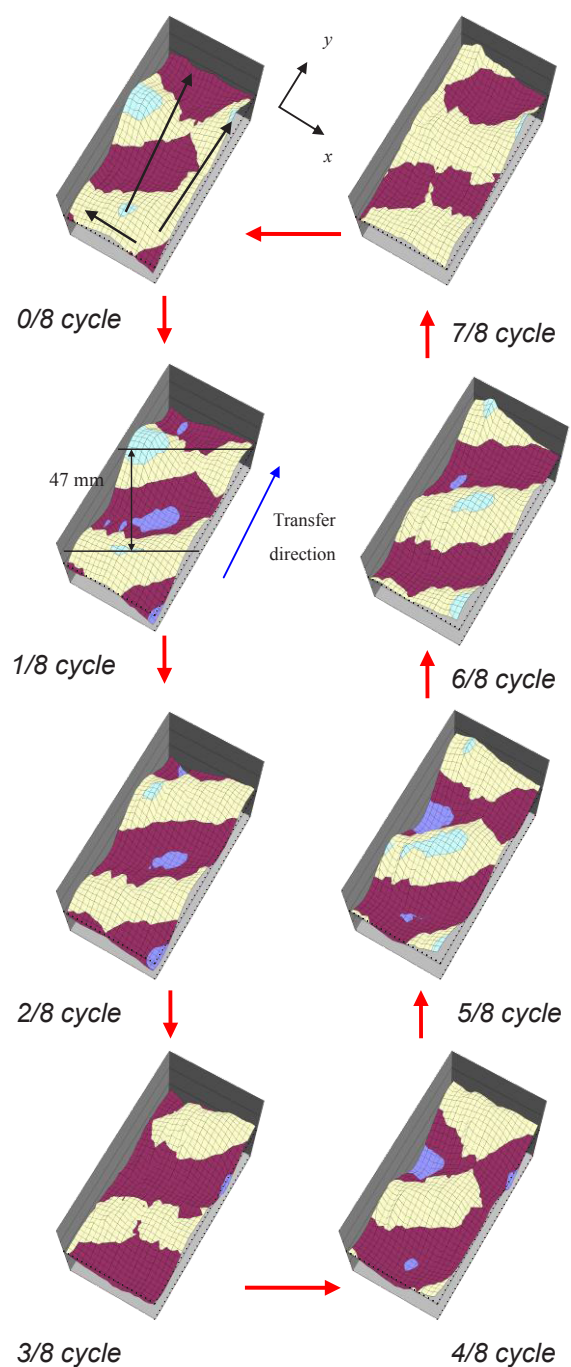


Figure 5. Transition of pressure loops. (Measurement range: 40×80 mm; vertical scale is ±400 Pa.)

transition of the pressure peaks generates the driving force.

Figure 6 shows distributions of vibration and pressure peaks. The positions of vibration peaks coincided with the pressure ones. However,

pressure peaks were observed even though the vibration peak was not observed at some places. Because the air flow is continuous, the air flow from the surrounding area into the center.

Translation Mechanism

When the carried plate contacts the vibration plate, the carried plate moves the reverse direction to the traveling wave. Therefore, the carried plate was completely levitated above the vibration plate, and traveling wave of the pressure affected the driving force to the carried plate. Many vortexes of the air flow were generated on the vibration plate. They synchronously rotated at the center and on the diagonal lines. Combinations of some vortexes generated air stream, which caused the driving force of the carried plate.

In an impact test, a vibration energy of 91% was reflected at the edge of the vibration plate. This caused the complex vibration mode. Some vortexes arranged in a row contributed to generate the air flow so that the driving force was generated.

CONCLUSIONS

In this paper, the plate behavior in two-dimensional transfer method using the ultrasonic levitation and the traveling wave was measured by focusing the generated pressure. Conclusions can be drawn as follows.

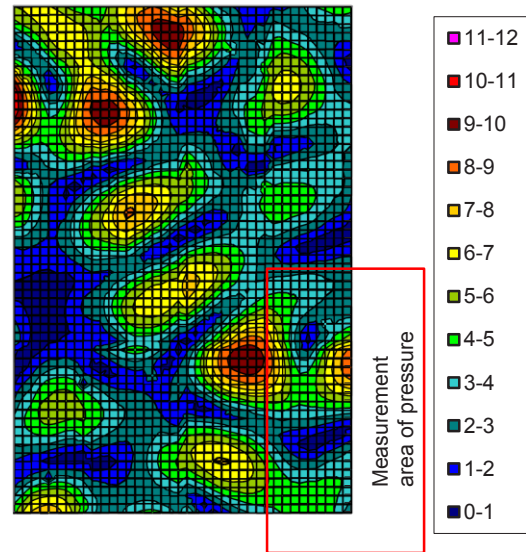
- The traveling waves of the plate vibration and pressure were observed.
- The air stream was generated between the source and sink vibrators by the combination of vortexes.

ACKNOWLEDGEMENTS

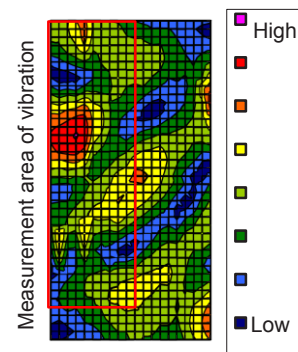
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(a) Vibration (80×120 mm).



(b) Pressure (40×80 mm).

Figure 6. Comparison of vibration and pressure.

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Surface Damage Growth Mitigation on KDP/DKDP Optics Using Single-Crystal Diamond Micro-Machining Ball End Mill Contouring

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INTRODUCTION

Weaknesses in the surface structure of an optic may result in damage initiated by exposure to high power laser energy. A subset of these laser-initiated surface damage sites will grow with continued laser exposure. These growing damage sites will negatively affect the life of the optic. A process to stabilize laser-initiated surface damage on KDP/DKDP optics by micro-machine contouring using a single-crystal diamond ball nose end mill has been shown to mitigate damage growth for subsequent laser exposure and thus extend optic life. The design of a machine to execute this process is presented. Particular attention is focused on the unique flexure design of the Mitigation Cartridge of the machine. The Mitigation Cartridge is the three axes subassembly that generates the surface damage mitigation contour. Other elements of the machine position the optic and monitor the process. The KDP/DKDP optic material is brittle. Parameters required for ductile mode machining and requirements for successful contour generation are presented along with how the machine elements meet these requirements.

OVERVIEW OF THE DESIGN

The Crystal Mitigation System (CMS) is made up of the Crystal Mitigation Machine and other components that are used in the mitigation process. The term Crystal, in this context, refers to the KDP/DKDP optics used in the National Ignition Facility (NIF) [1].

The Crystal Mitigation Machine, shown in Figures 1 and 2, has seven axes. Two axes, X and Y, position the Crystal's surface damage site over the mitigation tool. Two additional axes are used to focus the process monitoring camera and the inspection camera. The remaining three axes are used to perform the mitigation, see Figure 3.

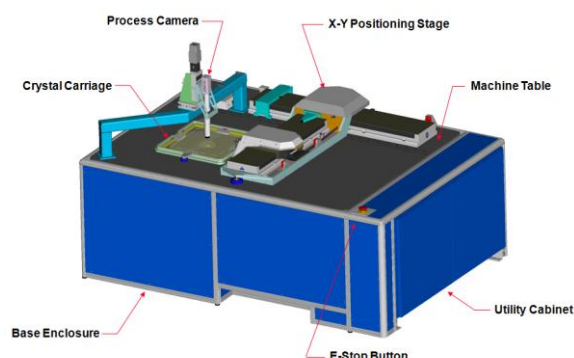


FIGURE 1. Model of Crystal Mitigation Machine showing elements of the machine.

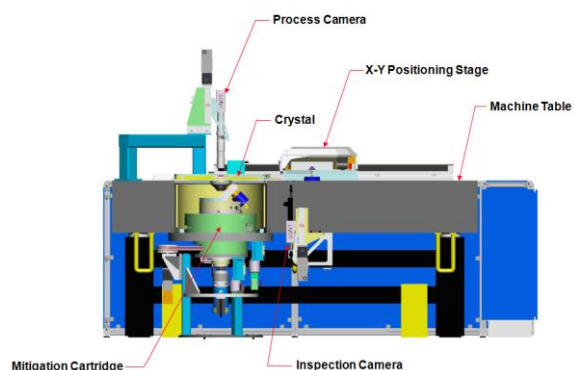


FIGURE 2. Model of Crystal Mitigation Machine showing elements of the machine in cross section.

Mitigation is the material removal process where a diamond tool is used to contour the surface of the optic, removing the fractures of the damage site, and leaving behind a surface feature that is benign to the laser system. The configuration of this aspect of the machine is similar to a vertical lathe where the tool is positioned radially (R-Axis) with respect to rotational axis (C-Axis), the axis of contour symmetry, as well as along the rotational axis (Z-Axis). But unlike a vertical lathe the workpiece is held in place and the tool is rotated about the C-Axis. The diamond tool is

also active, a ball end mill rotating at about 80,000 rpm.

The mitigation contour features are about 1 millimeter in diameter and 100 microns deep.

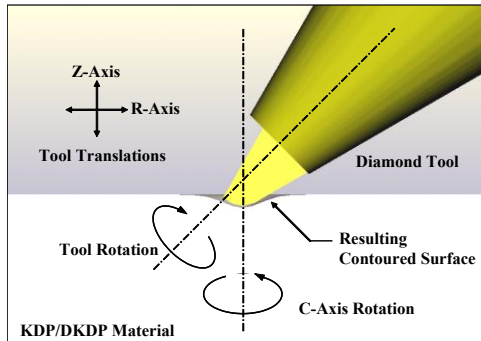


FIGURE 3. The mitigation axes. This figure shows the tool approaching the workpiece from above. In the machine, the tool approaches from below.

MITIGATION CARTRIDGE DESIGN

The three machine axes that perform the mitigation function are constructed within a subassembly called the Mitigation Cartridge. This subassembly was designed as a cartridge to facilitate machine repair and minimize downtime. The cartridge concept also aided in design risk minimization as most of the machine's technical risk was in mitigation function, the generation of the contoured surface. The Mitigation Cartridge completed prototype testing before the rest of the machine was designed.

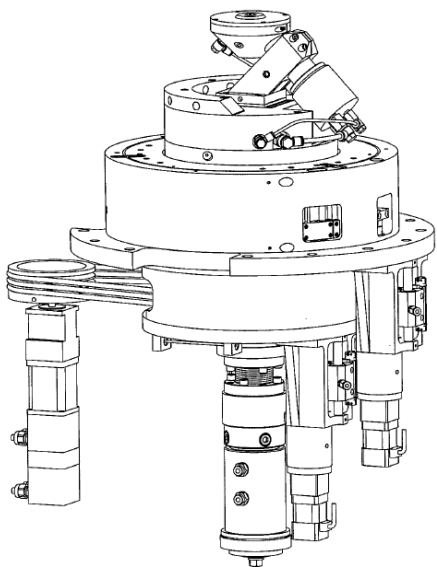


FIGURE 4. The Mitigation Cartridge.

The Crystal material, KDP/DKDP, is brittle. To be effective, the mitigation contour must be generated under the ductile machining regime [2]. This requires repeatability in tool position to insure the uniform chip thickness needed for ductile machining. Absolute positioning is much less important. The target chip thickness was determined based on knowledge gained from the diamond turning process used during Crystal production [3] with allowances made for differences in tool geometry and cutting parameters.

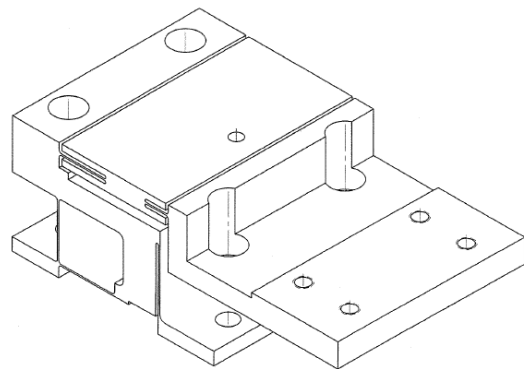


FIGURE 5. The flexure that converts vertical translation into horizontal translation for the R-Axis flexure linkage.

The cartridge was designed such that the electrical system (motor axis actuators and position feedback) are mounted on the fixed side of the C-Axis. This was to eliminate the need for an electrical slip ring. The tool spindle uses air bearings and an air turbine drive so only a gas rotary feedthrough is required.

With the exception of the motor actuators, there are no conventional bearings in the R-Axis or Z-Axis. The Z-Axis is made up completely of flexure linkage. The R-Axis is made up of flexure linkage with a small air bearing spindle to make the jump from the fixed to the rotating side of the C-Axis. Flexures were chosen over conventional bearings to achieve the repeatability requirements of the tool position for ductile machining.

The use of flexures resulted in a lower loop stiffness for the machine. This reduction in stiffness is insignificant relative to the Crystal stiffness. The tool position relative to the Crystal is dominated by the Crystal stiffness. The Crystals have a high aspect ratio (~40:1) and

are only supported on the Crystals edge when load into the machine.

The range of travels are 2 millimeters and 19 millimeters for the R-Axis and Z-Axis respectively. The C-Axis uses a commercially available air bearing spindle and a belt drive system.

MOTIVATION FOR DEVELOPING MACHINE

The National Ignition Facility relies on the KDP/DKDP optics to convert the laser frequency as it enters the target chamber. Surface damage sites may initiate as the high power laser passes through the Crystals. Some of the initiated damage sites will grow with subsequent laser exposure [4].



FIGURE 6. A NIF Crystal in an inspection frame.

If left unchecked, growing damage sites will limit the life of the Crystal. The damage site area may act as a shadow, or obscuration, in the laser path resulting in energy loss to the target. The damage site may also act as a phase object resulting in laser intensification and possibly damage downstream optics. Also, if the damage site grows too big, the depth of the damage may make it impossible to refinish the optic for additional use.

From a laser operations point of view, it is important to reduce the number of surface damage initiations that occur on these optics. There is an ongoing effort to do just that. Great strides have been made, in the quality of the material and the finishing process, that reduce the occurrences of these surface damage initiations but at the same time, the laser energy levels are also being pushed higher and higher.

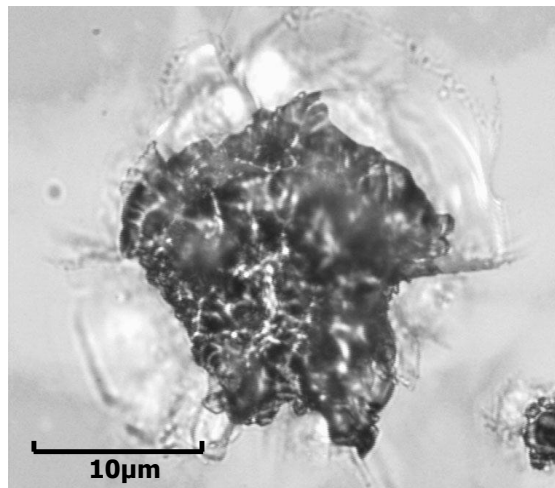


FIGURE 7. A damage site on the surface of a Crystal.

The mitigation process was developed to address this issue [5]. By mitigating the damage sites while they are still small, the optic life can be extended tenfold. To this end, an optic mitigation facility was designed, built, and integrated into NIF operations to process the Crystal as well as other types of optics. The Crystal Mitigation Machine is one of the pieces of equipment build for this facility. There are currently two Crystal Mitigation Machines in the facility.

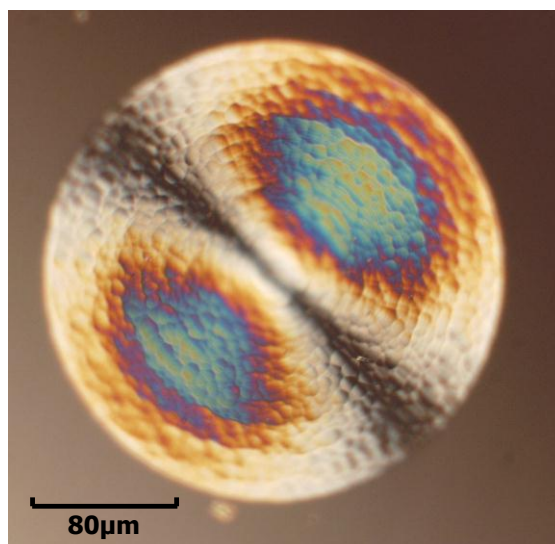


FIGURE 8. Mitigated site on the surface of a Crystal.



FIGURE 9. Crystal Mitigation Machine in production facility mitigating a surface damage site near the corner of the clear aperture of a NIF KDP/DKDP optic.

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BALL BEARING ANALYZER APPARATUS

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INTRODUCTION

This article describes an instrument to measure the error motion of rolling element bearings intended for precision applications. In the design presented here, an air bearing reference spindle with error motion of less than 10 nm rotates the inner race of the bearing under test. Non-influencing couplings are used to prevent rotation of the bearing outer race and apply an axial preload without distorting the bearing or influencing the measurement. Capacitive displacement sensors with 2 nm resolution target the non-rotating outer race. The error motion measurement repeatability is shown to be less than 25 nm.

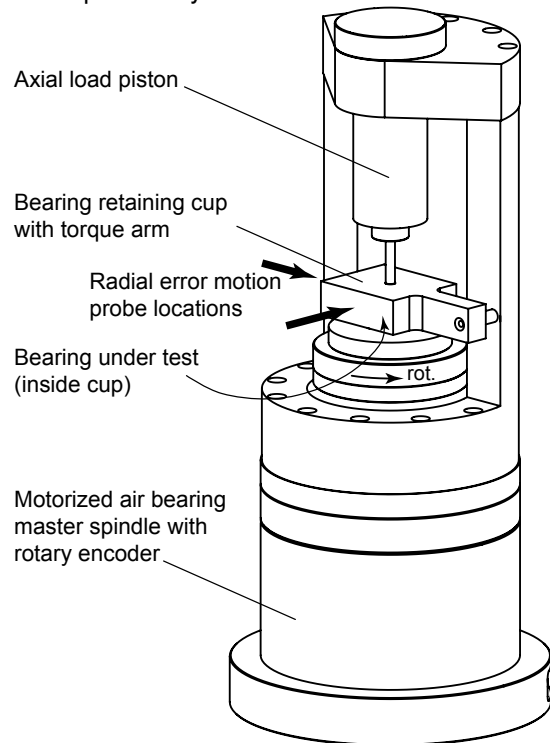


FIGURE 1. Bearing Analyzer Instrument

BACKGROUND

Manufacturers of rolling element bearings have improved performance over time as a result of ongoing metallurgical, tribological and manufactur-

ing research and development. However, one of the remaining challenges in rolling element bearing applications is measurement of rotational accuracy. This article describes an apparatus (Figure 1) that satisfies this requirement for most applications. Designers of precision machines may desire to characterize the performance of bearings in order to select the best bearings from a population or to select between similar bearings from different manufacturers. The full error motion of a bearing consists of movement in the five rigid body degrees of freedom other than pure rotation. These five components of error motion are described with radial, tilt (angular), and axial measurements. However, most applications use at least two bearings to provide tilt stiffness. Accordingly, the radial and axial error motion components shown in Figure 2 are most relevant. These are the error motion components measured by the instrument described here.

First, an axial preload must be applied to seat the rolling elements in the bearing races. Second, one of the races must spin under the influence of an applied torque. Third, rotation of the remaining race must be prevented in a way that leaves the radial, axial/face, and tilt displacements free to move. Finally, the bearing must be fixtured and measured without introducing off-axis loading or other distorting influences.

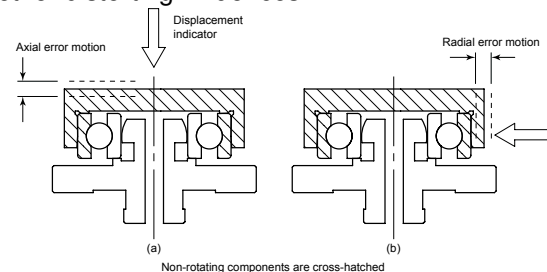


FIGURE 2. Measurement directions

Measurements made on the bearing under test reflect the combined contributions of errors in the

bearing and the errors of the reference spindle. Figure 2 shows the decomposition of the measurement separating these errors. The left-hand column of polar plots shows the synchronous and asynchronous components of a single radial measurement. The synchronous component is the repeatable path calculated by averaging several consecutive revolutions. Asynchronous error motion is the remaining error representing the revolution-by-revolution deviation from the synchronous component. Rolling element bearings generally have proportionally greater asynchronous error motion because of the non-integer relationship between input shaft speed and rolling element rotation. For example, the cage rotation of a 6204 ball bearing is 38.5% of the inner race rotation.

An error separation method may be applied to distinguish the synchronous error motion of the ball bearing under test from the reference air bearing spindle error. However, the synchronous error of the reference spindle is small compared to most rolling element bearings and is ignored in the results presented here. This is also true of the asynchronous component, which is about 1 nm for the reference spindle while typically 100 nm for rolling element bearings. Asynchronous error motion presents an additional challenge to the metrologist in that not all of the apparent motion between the displacement sensor and target should be assigned to the bearing under test because of test stand vibration, environmental influences, and other instrumentation or sensor noise. One of the biggest challenges in accurately measuring ball bearings is minimizing the contribution of external effects so that a reliable reading of asynchronous error is achieved.

For precision ball bearing analysis, careful design of structural and measurement loops is required for increased repeatability and accuracy. The computer-controlled bearing analyzer developed for this work is designed, manufactured, and programmed to take reliable error motion measurements with an accuracy of one microinch (25 nanometers).

BEARING ANALYZER HARDWARE AND COMPONENTS

The bearing analyzer spindle is a Professional Instruments motorized 3R Blockhead with brushless permanent magnet synchronous motor. A Heidenhain ERO 1221 (1024 counts/rev) rotary encoder is attached to the base of the spindle.

Lion Precision capacitance gages with 50 V/0.001 in. sensitivities record radial displacements.

The outer race of the bearing being tested is fitted into a precision-made holder; the inner race is mounted on a spherical pilot attached to the motorized 3R. During a test, the inner race rotates while the outer race is restricted from rotation. The air bearing load cell imparts axial preload. Two capacitance gages (not shown) are then set to gage on the ground outer surface of the bearing holder in X and Y directions¹.

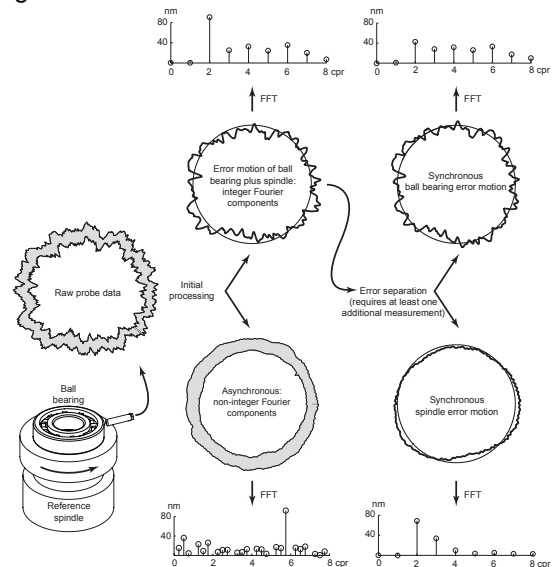


FIGURE 3. Bearing error motion analysis steps

DATA ACQUISITION AND ANALYSIS

The data acquisition is computer-controlled and can sample fast enough for analysis at 10,000 RPM. The data acquisition and analysis is performed by a stand-alone program written in the Matlab language. After the raw data is acquired, the user has an option to run a digital, window-type low pass filter with a linear phase. The user may specify the number of filter coefficients to be used and the cutoff (in upr). For fixed sensitive direction measurement, the fundamental frequency (1 upr) is removed from the data and the data is scaled to the appropriate units. The rotating sensitive direction measurement mathematics is described in ANSI B89.3.4M Axis of Rotation. Axial tests require that the fundamental frequency remain so the data is simply scaled to appropriate units and displayed.

CONCLUSION

The instrument described in this article can quantify the primary components of error motion of a

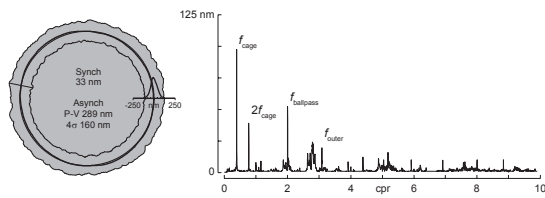


FIGURE 4. *Results of analysis*

rolling element bearing. The results are repeatable to the 100 nm level and may be used to identify bearing characteristics without the influence of the dynamics of a larger, compliant environment. Sample tests under varying axial load show that the error motion results are stable once sufficient load is applied to properly seat the rolling elements in the raceways. Additional testing of the effective contact angle shows that a significantly improved estimate of bearing geometry is obtained by comparing the predicted bearing fault frequencies with the measured values. This is important in contact angle calculations because without multiple spectral peaks to compare, the contact angle will be inaccurate.

EXPERIMENTAL APPARATUS FOR METROLOGY OF PARTIAL CYLINDERS

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INTRODUCTION

This paper presents a apparatus for the measurement of partial cylinders, and a method for removing tilt and radial alignment errors from the measurement data. The precision measurement of partial cylinders can be an important application. A number of issues are introduced by measuring a partial cylinder as opposed to a complete cylinder as is much more common. In particular, radial and tilt alignment errors of a partial cylinder are difficult to differentiate from form errors. This paper summarizes an analysis of the cylindricity instrument's sensitivity to geometry and alignment errors. The analysis is carried out using Homogeneous Transformation Matrices (HTM). By locating a Cartesian coordinate system at each point of interest in the structure it is possible to propagate the sensor/cylinder contact point back to a global reference while including the contribution of geometrical errors. Spindle errors are neglected in the study as they are small compared to other contributions (25 nanometers or less).

APPROACH

The apparatus is shown in 1. The main element of the apparatus is an air bearing rotary stage produced by Professional instruments. The stage is driven by a permanent magnet synchronous motor and the angular position is fed back from a sine wave optical encoder. The motion is controlled by an Aerotech Soloist precision motion control system. Form measurements are performed by a capacitance probe or a Lion Precision Air Bearing Capacitive Linear Variable Displacement Transducer (C-LVDT) The motion control and data acquisition are performed by a custom Matlab program using a National Instruments data acquisition system.

A partial cylinder to be measured is mounted in fixturing on the top of the stage. The mea-

surements are taken from scans that avoid measuring the edge of the partial cylinder to avoid the need to remove gross distortion from the data. The measurement probe is mounted in a Z axis stage, which is controlled manually between scans. In this case, a Coordinate Measuring Machine (CMM) was used as the Z axis.

The control PC runs a Mathworks Matlab program which communicates with the Aerotech Soloist using the Matlab Instrument Control Toolbox. The software on the Soloist drive allows Aerobasic programs to be sent asynchronously over an TCP/IP connection. The measurement parameters are set up using these commands, and then the Matlab program commands the Soloist to execute an Aerobasic program that is resident on the drive. This program performs the motion control, and also commands the drive I/ O circuitry to produce digital signals for triggering and synchronization of the data acquisition.

The Matlab program performs data acquisition using the Matlab Data Acquisition toolbox. The data acquisition is set up to require a trigger signal to start the scan and a clock signal to start each analog/ digital conversion. This allows precise synchronization between the position on the part to be measured and the measured data. The Aerobasic program on the Soloist generates the trigger signal on a digital I/O port, and clock signal is provided by the auxiliary encoder output. The auxiliary encoder output is driven by an encoder multiplier, which divides the signal from the feedback encoder by 256.

Workpiece HTMs

The global reference coordinate system is placed on the spindle's axis of rotation at the height of the workpiece mounting plate. Its orientation remains fixed with the stator and does not rotate with the rotor/workpiece. Additional coordinate systems

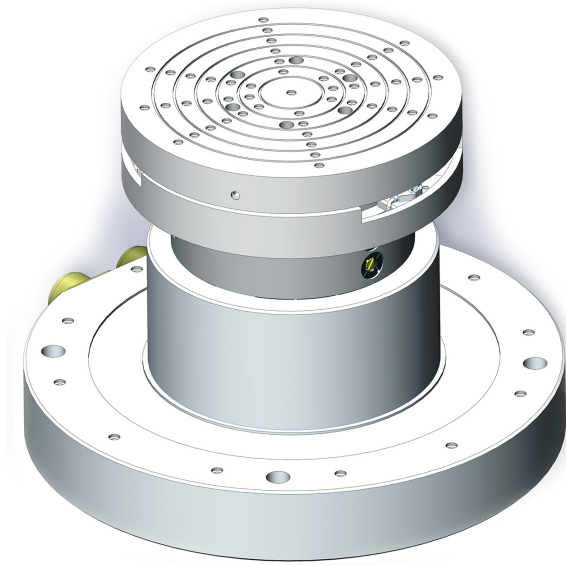


FIGURE 1. Cylindricity instrument.

are placed at points of interest such as linear and rotary bearings and contact points.

The next coordinate system in the sequence from global reference to the workpiece/sensor contact point shares the same origin as the global reference but rotates with the spindle rotor (angle γ). The next system incorporates imperfect centering of the workpiece on the spindle (eccentricity a_x and a_y). The cylinder and its fixture may also have wedge with respect to the spindle's axis of rotation (small tilt angles α_x and α_y). Finally, a contact point location on the cylinder may be specified in local coordinates.

$$\begin{Bmatrix} x \\ y \\ z \\ 1 \end{Bmatrix} = \begin{bmatrix} \cos \gamma & -\sin \gamma & 0 & 0 \\ \sin \gamma & \cos \gamma & 0 & 0 \\ 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 1 \end{bmatrix} \begin{bmatrix} 1 & 0 & 0 & a_x \\ 0 & 1 & 0 & a_y \\ 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 1 \end{bmatrix} \begin{bmatrix} 1 & 0 & \alpha_y & 0 \\ 0 & 1 & -\alpha_x & 0 \\ -\alpha_y & \alpha_x & 1 & 0 \\ 0 & 0 & 0 & 1 \end{bmatrix} \begin{Bmatrix} R \cos \beta \\ R \sin \beta \\ H \\ 1 \end{Bmatrix}$$

A location on the cylinder's centerline at height H is given by:

$$\begin{Bmatrix} x \\ y \\ z \end{Bmatrix}_{\text{CL}}^{\text{cyl}} = \begin{Bmatrix} (a_x + \alpha_y H) \cos \gamma + (-a_y + \alpha_x H) \sin \gamma \\ (a_y - \alpha_x H) \cos \gamma + (a_x + \alpha_y H) \sin \gamma \\ H \end{Bmatrix}$$

The point of contact between the cylinder of radius R and the stylus occurs at some angle β on the cylinder. Note: β is defined such that $\beta = 0$ in

the middle of the arc of the cylinder:

$$\begin{Bmatrix} x \\ y \\ z \end{Bmatrix}_{\text{pt}}^{\text{cyl}} = \begin{Bmatrix} (a_x + \alpha_y H) \cos \gamma + R \cos(\gamma + \beta) \\ (-a_y + \alpha_x H) \sin \gamma + (a_x + \alpha_y H) \sin \gamma \\ + R \sin(\gamma + \beta) \\ H - \alpha_y R \cos \beta - \alpha_x R \sin \beta \end{Bmatrix}$$

Sensor HTMs

The same procedure may be applied on the sensor side of the instrument. As before, we begin at the global coordinate reference frame on the spindle centerline. The first sensor-side coordinate frame is found directly above the global reference at the sensor height h . Ideally this point is on the sensor axis. However, we allow for error in the positioning of the sensor in the XY plane (sensor offset p_x and p_y). Furthermore, the sensor may be tilted about the Y and Z axes (sensor tilt ϕ_y and ϕ_z).

The nominal distance from the centerline of the spindle to the center of the ruby sphere maybe written as the difference between the cylinder's radius R and the spherical radius r plus an additional distance d . While d cannot be measured as an absolute distance by the capacitive sensor, any changes in d are detected with nanometer resolution. The total distance along the sensor axis is therefore $R - r + d$. The stylus of the measurement sensor will be nominally parallel with the sensor axis, but the center of the ruby sphere may not fall exactly on this axis (sphere offset c_y and c_z).

$$\begin{Bmatrix} x \\ y \\ z \\ 1 \end{Bmatrix} = \begin{bmatrix} 1 & 0 & 0 & 0 \\ 0 & 1 & 0 & 0 \\ 0 & 0 & 1 & h \\ 0 & 0 & 0 & 1 \end{bmatrix} \begin{bmatrix} 1 & 0 & 0 & p_x \\ 0 & 1 & 0 & p_y \\ 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 1 \end{bmatrix} \begin{bmatrix} 1 & -\phi_z & \phi_y & 0 \\ \phi_z & 1 & 0 & 0 \\ -\phi_y & 0 & 1 & 0 \\ 0 & 0 & 0 & 1 \end{bmatrix} \begin{bmatrix} 1 & 0 & 0 & R - r + d \\ 0 & 1 & 0 & 0 \\ 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 1 \end{bmatrix} \begin{bmatrix} r \cos \mu \cos \nu \\ r \sin \mu \cos \nu \\ r \sin \mu \sin \nu \\ 1 \end{bmatrix}$$

The location of the sphere's center with respect to the global reference frame is:

$$\begin{Bmatrix} x \\ y \\ z \end{Bmatrix}_0^{\text{sph}} = \begin{Bmatrix} R - r + d + p_x + \phi_y c_z - \phi_z c_y \\ p_y + c_y + \phi_z (R - r + d) \\ h + c_z - \phi_y (R - r + d) \end{Bmatrix}$$

It is convenient to specify the contact point of the sphere on the cylindrical workpiece with respect to the sphere's center using spherical coordinates.

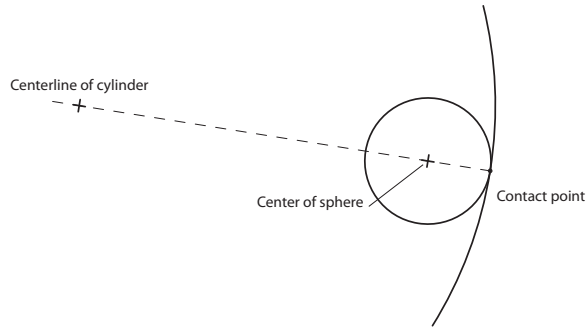


FIGURE 2. Sketch showing common normal shared by the cylinder (workpiece) and sphere (sensor stylus).

$$\begin{Bmatrix} x \\ y \\ z \end{Bmatrix}_{pt}^{sph} = \begin{Bmatrix} R - r + d + p_x + \phi_y c_z - \phi_z c_y + r \cos \mu \cos \nu \\ -\phi_z r \cos \nu \sin \mu + \phi_y r \sin \nu \\ p_y + c_y + \phi_z (R - r + d) + \phi_z r \cos \mu \cos \nu \\ + r \cos \nu \sin \mu \\ h + c_z - \phi_y (R - r + d) - \phi_y r \cos \mu \cos \nu \\ + r \sin \nu \end{Bmatrix}$$

Solving for the point of contact

Once the geometry of both the instrument and its errors are defined five unknowns remain to be determined:

- β = angle at the point of contact
- ν = azimuthal angle at POC
- μ = altitudinal angle at POC
- d = distance to sphere's center
- H = height at point of contact

Three equations may be obtained by equating the point of contact on the workpiece side (cylinder) to the point of contact on the sensor side (ruby sphere).

$$\begin{Bmatrix} x \\ y \\ z \end{Bmatrix}_{pt}^{cyl} = \begin{Bmatrix} x \\ y \\ z \end{Bmatrix}_{pt}^{sph}$$

Three more equations are found by recognizing that the sphere and cylinder must always share a common normal vector. The lengths of the vectors are nominally related to the difference in radii, but the stackup of geometry errors introduces a sixth unknown (ψ).

$$\begin{Bmatrix} x \\ y \\ z \end{Bmatrix}_{CL}^{cyl} - \begin{Bmatrix} x \\ y \\ z \end{Bmatrix}_{pt}^{cyl} = \psi \left(\begin{Bmatrix} x \\ y \\ z \end{Bmatrix}_{pt}^{sph} - \begin{Bmatrix} x \\ y \\ z \end{Bmatrix}_0^{sph} \right)$$

This system of equations is quite nonlinear, but a few simplifying assumptions may be made. The

altitudinal (ν) and azimuthal (μ) angles of the sphere's contact point may be treated as small angles. Furthermore, the point of contact will remain near the XZ plane, regardless of the spindle's rotation (γ). While both γ and β will be large angles, their difference δ will be small. Two transcendental terms remain; however, these are known functions of the measured spindle angle γ (that's good news).

The remaining nonlinearity precludes a closed form solution. However, the effect of individual geometry error terms may be estimated by setting the other terms to zero. By this approach, several of the error components are found to have no first-order contribution to the instrument error. The remaining terms contribute to error in the measured displacement by:

$$d = (\alpha_y H + a_x) \cos \gamma + (\alpha_x H - a_y) \sin \gamma$$

where:

- α = workpiece tilt
- ϕ = capacitive sensor tilt
- γ = spindle rotation angle—by encoder
- H = distance along centerline—by CMM

$$\begin{aligned} A_0 &= \sum_{n=2}^{\infty} a_n \frac{2\pi(n^2 - 1) \sin\left(\frac{\pi n}{2}\right) + 8n \cos\left(\frac{\pi n}{2}\right)}{n(n^2 - 1)(\pi^2 - 8)} \\ A_n &= \sum_{n=2}^{\infty} -a_n \frac{2(n^2 - 1) \sin\left(\frac{\pi n}{2}\right) + 4n\pi \cos\left(\frac{\pi n}{2}\right)}{n(n^2 - 1)(\pi^2 - 8)} \\ B_n &= \sum_{n=2}^{\infty} b_n \frac{4n \cos\left(\frac{\pi n}{2}\right)}{\pi - \pi n^2} \end{aligned}$$

$$\begin{aligned} A_0 &= \sum_{n=2}^{\infty} \frac{2a_n}{\pi^2 - 8} \left(\frac{\pi}{n} \sin\left(\frac{\pi n}{2}\right) + \frac{4}{n^2 - 1} \cos\left(\frac{\pi n}{2}\right) \right) \\ A_n &= \sum_{n=2}^{\infty} \frac{-2a_n}{\pi^2 - 8} \left(\frac{1}{n} \sin\left(\frac{\pi n}{2}\right) + \frac{2\pi}{n^2 - 1} \cos\left(\frac{\pi n}{2}\right) \right) \\ B_n &= \sum_{n=2}^{\infty} \frac{4nb_n}{\pi - \pi n^2} \cos\left(\frac{\pi n}{2}\right) \end{aligned}$$

$$\begin{aligned} A_0 &= \sum_{n=2}^{\infty} \frac{2a_n}{\pi^2 - 8} \left(\frac{\pi}{n} \sin \frac{\pi n}{2} + \frac{4}{n^2 - 1} \cos \frac{\pi n}{2} \right) \\ A_n &= \sum_{n=2}^{\infty} \frac{-2a_n}{\pi^2 - 8} \left(\frac{1}{n} \sin \frac{\pi n}{2} + \frac{2\pi}{n^2 - 1} \cos \frac{\pi n}{2} \right) \\ B_n &= \sum_{n=2}^{\infty} \frac{4nb_n}{\pi - \pi n^2} \cos \frac{\pi n}{2} \end{aligned}$$

CONCLUSIONS

An apparatus and analysis technique has been developed for determining the geometry of partial cylinders. The analysis technique yields results that are approximate but an optimal estimate may not be possible in the face of the unknowns inherent to the analysis.

PREDICTION AND EXPERIMENTAL VERIFICATION OF MOTION ERRORS IN POROUS AIR BEARING STAGES

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INSTRUCTIONS

This paper presents an estimation method of motion errors of porous air bearing stages and its experimental verification. The motion errors are predicted from the rail form errors in cooperation with the transfer function method. The rail form errors are measured using displacement sensors and specially designed measuring device. Then, the motion errors are predicted and verified from the experiment.

PREDICTION OF MOTION ERRORS USING THE TRANSFER FUNCTION METHOD

The transfer function method is basically used in predicting the motion errors of the precision stages [1]. The transfer function $K(\omega)$ is defined as the ratio of reaction force $f_e(\omega)$ acting on one bearing pad to magnitude of the sinusoidal rail form error $e(\omega)$ at a spatial frequency $\omega(=2\pi/\lambda)$ when the bearing pad moves on the rail without variation of motion, as shown in Figure 1. λ means a spatial period of the rail form error, δ is the magnitude of rail form error, h_0 is an average clearance of the bearing pad, and l is bearing length. The transfer function is a characteristic of the bearing, doesn't change unless the bearing model or geometric size change. The reaction force is calculated from the Hertz contact equation for rolling bearings, and it is calculated from the Darcy's law and Reynolds equation for porous air bearings [2, 3].

$$K(\omega) = f_e(\omega) / e(\omega) \quad (1)$$

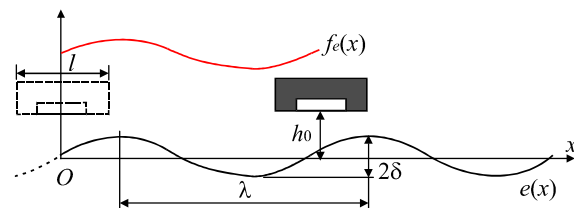


FIGURE 1. Definition of the transfer function

The precision stages are practically composed of identical several bearing pads. Therefore, geometrical relation between bearing pads is used to predict the motion errors after analyzing the transfer function for only one bearing pad without analyzing all bearing pads completely. This method has great advantages, that is, motion errors can be calculated by using the same algorithm regardless of bearing type (rolling bearing, air bearing, hydrostatic bearing, etc) if the transfer function for the bearing is known.

Figure 2 shows the model of air bearing table for predicting motion errors. When δ_z , δ_y , θ_x , θ_y , θ_z represent vertical, horizontal straightness errors, and roll, pitch and yaw errors at table center, respectively. Then, vertical displacement z_{ij} and horizontal displacement y_{ij} due to the rail form errors in the i , j th bearing are expressed by equation (2).

$$\begin{aligned} z_{ij} &= \delta_z + \theta_y X_{ci} - Y_{cj} \theta_x \\ y_{ij} &= \delta_y - X_{ci} \theta_z - z_{cj} \theta_x \end{aligned} \quad (2)$$

Where, X_{ci} and Y_{ci} are the distance between bearing center and table center in the x and y axis. z_c is the distance between the rail center and upper surface of table in the vertical direction. The equilibrium equations of force and moment on the 5 DOF directions are expressed by equation (3).

$$\begin{aligned} \sum_j^n \sum_i^m (f_{z,ij} - K_z z_{ij}) &= 0 \\ \sum_j^n \sum_i^m \{f_{z,ij} X_{ci} - K_z z_{ij} X_{ci}\} &= 0 \\ \sum_j^n \sum_i^m (f_{z,ij} Y_{cj} - K_z z_{ij} Y_{cj}) + z_c \sum_j^n \sum_i^m (f_{y,ij} - K_y y_{ij}) &= 0 \\ \sum_j^n \sum_i^m (f_{y,ij} - K_y y_{ij}) &= 0 \\ \sum_j^n \sum_i^m \{f_{y,ij} X_{ci} - K_y y_{ij} X_{ci}\} &= 0 \end{aligned} \quad (3)$$

$f_{z,ij}$ and $f_{y,ij}$ are reaction forces of the i, j th bearing in the z (vertical) and y (horizontal) directions. K_z and K_y are the stiffnesses of bearing in the z and y directions. m and n are the number of bearings in the x and y axis.

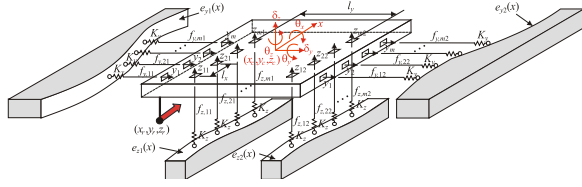


FIGURE 2. Model for analyzing the motion errors

Bearing reaction forces $f_{z,ij}$ and $f_{y,ij}$ are calculated using the transfer function and the rail form errors, as shown in equation (4). a_k and b_k are the Fourier coefficients of the rail form errors, and L is the rail length. Therefore, the 5 DOF motion errors ($\delta_z, \delta_y, \theta_x, \theta_y, \theta_z$) can be calculated by solving the equation (3).

$$e(x) = \sum_{k=1}^{\infty} \left(a_k \cos \frac{2k\pi}{L} x + b_k \sin \frac{2k\pi}{L} x \right) \quad (4)$$

$$f_e(x) = \sum_{k=1}^{\infty} K \left(\frac{2k\pi}{L} \right) \left(a_k \cos \frac{2k\pi}{L} x + b_k \sin \frac{2k\pi}{L} x \right)$$

MEASUREMENT OF RAIL FORM ERRORS

Procedure of predicting motion errors is as follows. Firstly, we calculate the transfer function from the geometrical information of a porous air bearing pad, and governing equations(Darcy's law and Reynolds equation). Then, the rail form errors are measured. Thirdly, motion errors are estimated from the rail form errors using the transfer function method. Finally, estimated motion errors are compared with experimented one.

Figure 3 shows the air bearing stage designed to predict motion errors. The stage is composed of opposite porous air bearings in vertical and horizontal directions.

To measure rail form errors $e_{z1}, e_{z2}, e_{z3}, e_{z4}, e_{y1}, e_{y2}$, we used precision displacement sensors and specially designed equipment to use measurement method based on the sequential two point method [4]. The measurement principle of rail form errors is shown in Figure 4. Horizontal rail form error $e_h(x)$ and horizontal linear motion error(straightness error) $\delta_h(x)$ of the measuring device(feed table) are calculated from sensor A(output signal: $R_A(x)$), sensor B(output signal: $R_B(x)$), and angular error $\theta_y(x)$ of the

measuring device, as shown in equation (5). Displacement difference between two abreast sensors A and B caused by angular error of the measuring device is compensated. The vertical rail form errors are also measured with the same method. If we use additional one sensor in opposite direction, parallelism error between two rails can be also measured. The measurement method is verified by comparing the two linear motion errors of the measuring device which are calculated from left and right rail form errors, respectively.

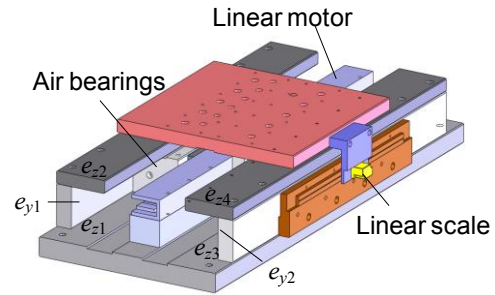


FIGURE 3. Air Bearing Stage

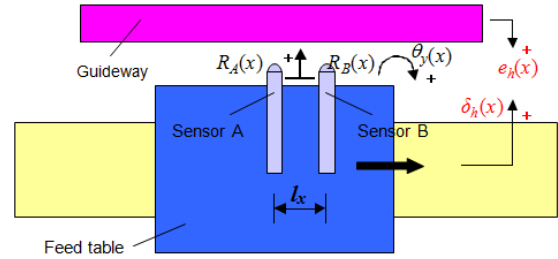


FIGURE 4. Measurement principle of rail form error

$$R_A(x_i) = e_h(x_i) + \delta_h(x_i)$$

$$R_B(x_i) = e_h(x_{i+1}) + \delta_h(x_i) - l_x \cdot \theta_y(x_i)$$

$$R_A(x_{i+1}) = e_h(x_{i+1}) + \delta_h(x_{i+1}) \quad (5)$$

$$R_B(x_{i+1}) = e_h(x_{i+2}) + \delta_h(x_{i+1}) - l_x \cdot \theta_y(x_{i+1})$$

$$e_h(x_{i+1}) = e_h(x_i) + R_B(x_i) - R_A(x_i) + l_x \cdot \theta_y(x_i)$$

$$\delta_h(x_{i+1}) = \delta_h(x_i) + R_A(x_{i+1}) - R_B(x_i) - l_x \cdot \theta_y(x_i)$$

$$e_h(x_0) = \delta_h(x_0) = 0$$

Figure 5 shows the measuring device designed. It is similar to normal porous air bearing table, and it has several displacement sensors to measure the rail form errors using the sequential two point method. Figure 6 shows the experiment setup. A step motor and a wire rope

system are used to move the measuring device with constant speed. Weight balance is installed to maintain constant tension of the wire rope. And a laser interferometer is used to measure angular error between two abreast displacement sensors.

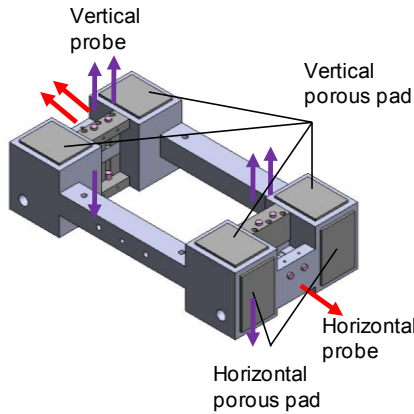


FIGURE 5. Measuring device for rail form error measurement

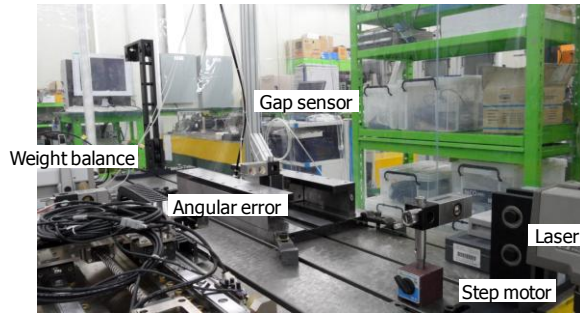


FIGURE 6. Experimental setup

Figure 7 shows measured rail form errors using the proposed method. The results show that amplitude of rail form errors for horizontal rails are $e_{y1}=4.7$ and $e_{y2}=1.1 \mu\text{m}$, while the rail form errors for vertical rails are $e_{z1}=1.3$, $e_{z2}=1.3$, $e_{z3}=2.2$, $e_{z4}=2.7 \mu\text{m}$, respectively.

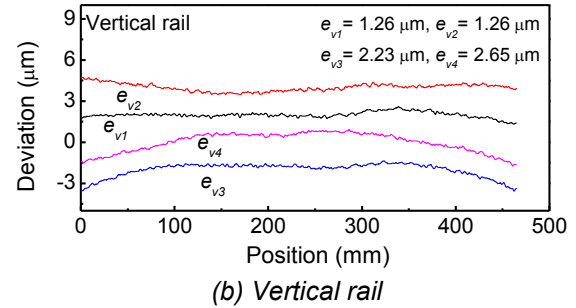
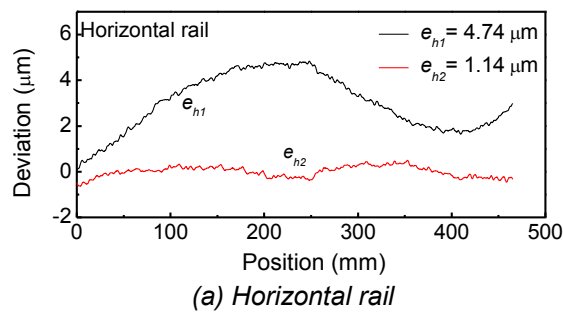
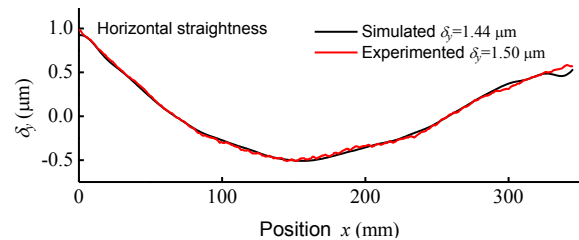


FIGURE 7. Measured rail profile errors

COMPARISON BETWEEN ESTIMATED AND MEASURED MOTION ERRORS

The experiment setup for measurement of motion errors is the same as that of the rail form error measurement. In this configuration, the measuring device takes a role as a moving table as well as rail measuring device. Therefore, additional table, linear motor and scale are not required. We can measure rail form errors and table linear motion errors simultaneously with this setup. After motion errors are estimated from the rail form errors, estimated and measured motion errors are compared. Horizontal and vertical straightness errors are measured by using two sequential point method, as shown in Eq (5). Yaw and pitch errors are measured with a laser interferometer. Roll error is measured with electronic level meter.

Figure 8 presents the comparison results between simulated and experimented data in horizontal straightness error and yaw error. The results show that simulated motion errors are very well coincided with the experiment. The reason for extremely good agreement is because we used measuring device as a moving table. Therefore, there are no critical error sources in measuring rail form errors and table motion errors. From this result, the estimation method of motion errors using the transfer function method is verified.



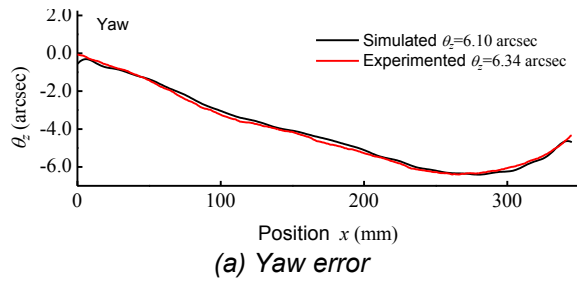


FIGURE 8. Comparison between simulated and experimented results

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A SPINDLE FOR A LARGE LIQUID MIRROR TELESCOPE

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ABSTRACT

Astronomers demand diffraction-limited optics, and when the optic is a liquid the requirement calls for pure rotation. Mercury-mirror telescopes need spindles with high load capacity, high stiffness, high accuracy, and speed control to a part-per-million. Balance and leveling are also crucial. Tilt stiffness is necessary to avoid a runaway condition where an off-axis load gains force if the mercury runs sideways. [1]



FIGURE 1. Artist's conception of liquid mirror astronomy. [2]



Here is an unnamed one. [3]

FIGURE 2. Despite being restricted to a strip of sky about the width of the apparent size of the moon, 100,000 galaxies are in range.



Figure 3. One of the largest telescopes in the world is this 6-meter one at the University of British Columbia. Finished in 2004, the cost was a fraction of that for a conventional telescope of this size. The mercury surface is deemed to be parabolic within 9 nanometers rms at 7 rpm. [4]



Figure 4. All structures in the load path must be stable, stiff, and symmetrical, including the spindle, the space-frame truss, and the 36 leveling screws connecting the stator to the pad.



Figure 5. The upper and lower thrust bearings have separate feed lines enabling equal air film thicknesses under the 3-ton load. The axis of rotation is tilted 13.1 arc-seconds to compensate for the Coriolis effect [5]. Air exhausting from the spindle carries away heat from the integrated frameless brushless servo motor. A frameless integrated 2,500 line count optical encoder provides square wave feedback to the phase-lock-loop controlled low noise linear amplifier.

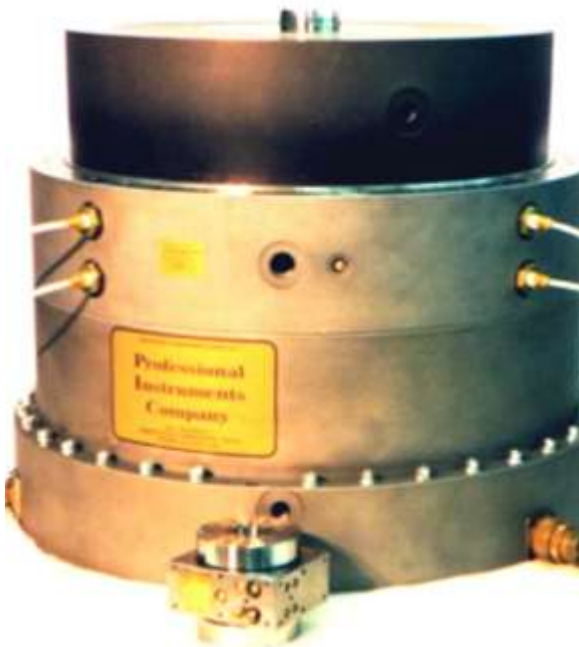


Figure 6. A 4B is dwarfed by the 23R. Axial load capacity is 400 pounds for the 4B and 16,000 pounds for the 23R. [6]

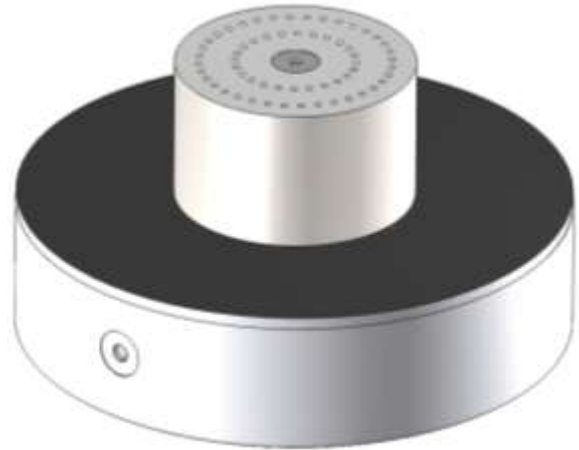


Figure 7. The enormous hydrostatic force of the air films requires extra-thick thrust plates and a solid shaft with dual bolt circles.



Figure 8. Roto-grinding a 700 pound thrust plate.



Figure 9. Boring the stator on a belt-driven 10-Inch Blockhead in our Matsuura machining center. A tilt base sets the axis of rotation parallel to the machine vertical travel. Fine infeed was done by moving the table in the non-sensitive direction.



Figure 10. Visual inspection by Sam Arneson.



Figure 11. A lift-off test for axial load capacity.



Figure 12. Assembling the mirror support segments.



Figure 13

An oxide layer forms promptly so that mercury fumes are only hazardous at first.

Conclusion

The Large Zenith Telescope confirms that large mercury mirrors of astronomical quality can be produced with a spindle and drive assembly supported by a robust structural loop from the rotor to the mirror surface and from the stator to bedrock.

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GEOMETRIC ERROR MODELING OF A SUPER-SIZE FLOOR TYPE BORING MACHINE

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ABSTRACT

Volumetric error analysis of a super-size floor type boring machine is conducted through geometric error modeling of the machine tool. Thermal deformation errors of the joint and shape elements are included in the model. Boring and milling errors are estimated through simulation.

INTRODUCTION

Super-size floor type boring machines are used for heavy-metal milling, boring, drilling and tapping on mega parts for the construction, oil field, wind energy, plant, shipbuilding industries. To make big machining volume, they are made of boring and milling spindle (W-axis) assembled on the ram (Z-axis). Machining accuracy depends upon deviation from the planned relative movement between the tool and the workpiece. For machine tools, this relative error is called the volumetric error. This is generated from joint errors of moving axes and shape deformations from the machine tool structures. Strokes of the X, Y, Z axes are over several meters. As the size is too big, machining error is bigger than small and medium size machine tools. It depends upon non-repeatable joint and shape deformation errors due to internal and external thermal error sources.

In this paper, to identify error behavior of the super-size floor type boring machine, homogeneous transformation matrix (HTM) and kinematic error chain [1-4] are applied for modeling the 3D geometric error. Thermal gains and time constants of thermal error models are formulated through first order systems. Milling and boring errors are estimated for error analysis.

VOLUMETRIC ERROR MODEL

Fig. 1 shows schematic diagram of a super-size floor type boring machine. There are X, Y, Z, V, W and B axes as joint elements. Floor bed, rotary table, column, head, ram, spindle, etc. are

shape elements. Geometric error model consists of several shape and joint error elements. The shape error is generated from structure of a machine tool. It is represented as a shape transformation element in the error model. The joint error occurs at a moving joint of a machine tool. The machine has five feed axes and one rotary axis. Rack and pinion for the X-axis, ballscrews in Y, Z, W and V axes, and B-axis are joint elements. Fig. 2 shows the kinematic chain of the machine tool. CS1 is the origin of the machine. CS2 is the coordinates at the pinion attached to the column. CS1-CS2, CS1-CS4, CS5-CS6, CS7-CS8, CS3-CS9, CS10-CS11, CS12-CS13, CS14-CS15 are shape transformation elements $S_{1-2}, S_{1-4}, S_{5-6}, S_{7-8}, S_{3-9}, S_{10-11}, S_{12-13}, S_{14-15}$, and CS3, CS5, CS7, CS10, CS12, CS14 are joint transformation elements $J_{2-3}, J_{4-5}, J_{6-7}, J_{9-10}, J_{11-12}, J_{13-14}$, respectively.

Ahn [2] and Kim [3-4] modeled volumetric errors in a three-axis machine tool by homogeneous transformation matrices considering geometric and thermal deformation errors. Contrary to the small machine tools, the super-sized floor type boring machine has longer height. Temperature difference between the floor and ceiling generates significant thermal deformation of the column. Following shows modeled shape and joint transformation elements:

$$[S_{1-2}]_{kneel} = \begin{bmatrix} 1 & -\varepsilon_{21}^x(T) & \varepsilon_{21}^y(T) & a_1 + \Delta a_1(T) \\ \varepsilon_{21}^x(T) & 1 & -\varepsilon_{21}^y(T) & b_1 + \Delta b_1(T) \\ -\varepsilon_{21}^y(T) & \varepsilon_{21}^x(T) & 1 & c_1 + \Delta c_1(T) \\ 0 & 0 & 0 & 1 \end{bmatrix} [S_{1-4}]_{kneel} = \begin{bmatrix} 1 & -\varepsilon_{42}^x(T) & \varepsilon_{42}^y(T) & a_2 + \Delta a_2(T) \\ \varepsilon_{42}^x(T) & 1 & -\varepsilon_{42}^y(T) & b_2 + \Delta b_2(T) \\ -\varepsilon_{42}^y(T) & \varepsilon_{42}^x(T) & 1 & c_2 + \Delta c_2(T) \\ 0 & 0 & 0 & 1 \end{bmatrix}$$

$$[S_{3-9}]_{kneel} = \begin{bmatrix} 1 & -\varepsilon_{93}^x(T) & \varepsilon_{93}^y(T) & a_3 + \Delta a_3(T) \\ \varepsilon_{93}^x(T) & 1 & -\varepsilon_{93}^y(T) & b_3 + \Delta b_3(T) \\ -\varepsilon_{93}^y(T) & \varepsilon_{93}^x(T) & 1 & c_3 + \Delta c_3(T) \\ 0 & 0 & 0 & 1 \end{bmatrix} [S_{5-6}]_{kneel} = \begin{bmatrix} 1 & -\varepsilon_{65}^x(T) & \varepsilon_{65}^y(T) & \Delta a_4(T) \\ \varepsilon_{65}^x(T) & 1 & -\varepsilon_{65}^y(T) & b_4 + \Delta b_4(T) \\ -\varepsilon_{65}^y(T) & \varepsilon_{65}^x(T) & 1 & \Delta c_4(T) \\ 0 & 0 & 0 & 1 \end{bmatrix}$$

$$[S_{7-8}]_{kneel} = \begin{bmatrix} 1 & -\varepsilon_{87}^x(T) & \varepsilon_{87}^y(T) & a_5 + \Delta a_5(T) \\ \varepsilon_{87}^x(T) & 1 & -\varepsilon_{87}^y(T) & b_5 + \Delta b_5(T) \\ -\varepsilon_{87}^y(T) & \varepsilon_{87}^x(T) & 1 & c_5 + \Delta c_5(T) \\ 0 & 0 & 0 & 1 \end{bmatrix} [S_{10-11}]_{kneel} = \begin{bmatrix} 1 & -\varepsilon_{1110}^x(T) & \varepsilon_{1110}^y(T) & a_6 + \Delta a_6(T) \\ \varepsilon_{1110}^x(T) & 1 & -\varepsilon_{1110}^y(T) & b_6 + \Delta b_6(T) \\ -\varepsilon_{1110}^y(T) & \varepsilon_{1110}^x(T) & 1 & c_6 + \Delta c_6(T) \\ 0 & 0 & 0 & 1 \end{bmatrix}$$

$$\begin{aligned}
[S_{2-3}]_{actual} &= \begin{bmatrix} 1 & -\varepsilon_{z1}^s(T) & \varepsilon_{y1}^s(T) & \Delta a_1(T) \\ \varepsilon_{z1}^s(T) & 1 & -\varepsilon_{y1}^s(T) & b_1 + \Delta b_1(T) \\ -\varepsilon_{y1}^s(T) & \varepsilon_{z1}^s(T) & 1 & c_1 + \Delta c_1(T) \\ 0 & 0 & 0 & 1 \end{bmatrix} [S_{4-5}]_{actual} = \begin{bmatrix} 1 & -\varepsilon_{z8}^s(T) & \varepsilon_{y8}^s(T) & \Delta a_8(T) \\ \varepsilon_{z8}^s(T) & 1 & -\varepsilon_{y8}^s(T) & b_8 + \Delta b_8(T) \\ -\varepsilon_{y8}^s(T) & \varepsilon_{z8}^s(T) & 1 & c_8 + \Delta c_8(T) \\ 0 & 0 & 0 & 1 \end{bmatrix} \\
[J_{2-3}]_{actual} &= \begin{bmatrix} 1 & -\varepsilon_{x1}(x,T) & \varepsilon_{y1}(x,T) & x + \delta_x(x,T) + P_x(T) \cdot x + S_{xy} \cdot x \\ \varepsilon_{x1}(x,T) & 1 & -\varepsilon_{y1}(x,T) & \delta_y(x,T) \\ -\varepsilon_{y1}(x,T) & \varepsilon_{x1}(x,T) & 1 & \delta_z(x,T) \\ 0 & 0 & 0 & 1 \end{bmatrix} \\
[J_{4-5}]_{actual} &= \begin{bmatrix} 1 & -\varepsilon_{z3}(v,T) & \varepsilon_{y3}(v,T) & \delta_x(v,T) - S_{xz} \cdot v \\ \varepsilon_{z3}(v,T) & 1 & -\varepsilon_{y3}(v,T) & \delta_y(v,T) \\ -\varepsilon_{y3}(v,T) & \varepsilon_{z3}(v,T) & 1 & v + \delta_z(v,T) + P_z(T) \cdot v \\ 0 & 0 & 0 & 1 \end{bmatrix} \\
[J_{6-7}]_{actual} &= \begin{bmatrix} 1 & -\varepsilon_{z3}(r,T) & \varepsilon_{y3}(r,T) & \delta_x(r,T) \\ \varepsilon_{z3}(r,T) & 1 & -\varepsilon_{y3}(r,T) & \delta_y(r,T) \\ -\varepsilon_{y3}(r,T) & \varepsilon_{z3}(r,T) & 1 & \delta_z(r,T) \\ 0 & 0 & 0 & 1 \end{bmatrix} \\
[J_{9-10}]_{actual} &= \begin{bmatrix} 1 & -\varepsilon_{z4}(y,T) & \varepsilon_{y4}(y,T) & \delta_x(y,T) + S_{xy} \cdot y \\ \varepsilon_{z4}(y,T) & 1 & -\varepsilon_{y4}(y,T) & y + \delta_y(y,T) + P_y(T) \cdot y \\ -\varepsilon_{y4}(y,T) & \varepsilon_{z4}(y,T) & 1 & \delta_z(y,T) \\ 0 & 0 & 0 & 1 \end{bmatrix} \\
[J_{11-12}]_{actual} &= \begin{bmatrix} 1 & -\varepsilon_{z5}(w,T) & \varepsilon_{y5}(w,T) & \delta_x(w,T) \\ \varepsilon_{z5}(w,T) & 1 & -\varepsilon_{y5}(w,T) & \delta_y(w,T) - S_{wy} \cdot w \\ -\varepsilon_{y5}(w,T) & \varepsilon_{z5}(w,T) & 1 & w + \delta_z(w,T) + P_z(T) \cdot w \\ 0 & 0 & 0 & 1 \end{bmatrix} \\
[J_{13-14}]_{actual} &= \begin{bmatrix} 1 & -\varepsilon_{z6}(z,T) & \varepsilon_{y6}(z,T) & \delta_x(z,T) \\ \varepsilon_{z6}(z,T) & 1 & -\varepsilon_{y6}(z,T) & \delta_y(z,T) - S_{zy} \cdot z \\ -\varepsilon_{y6}(z,T) & \varepsilon_{z6}(z,T) & 1 & z + \delta_z(z,T) + P_z(T) \cdot z \\ 0 & 0 & 0 & 1 \end{bmatrix} \quad (1)
\end{aligned}$$

TABLE 1. Definition of Coordinate Systems.

Coordinates	Location
CS1	Reference coordinate
CS2	X-axis rack
CS3	X-axis pinion center
CS4	V-axis ball screw bearing center
CS5	Ball screw nut center of rotary table
CS6	B-axis spindle center of rotary table
CS7	Center of rotary table
CS8	Origin of workspace of machine
CS9	Y-axis ball screw bearing center
CS10	Ball screw nut center of head
CS11	W-axis ball screw bearing center
CS12	Ball screw nut center of ram
CS13	Z-axis ball screw bearing center
CS14	Ball screw nut center of spindle
CS15	Tool grasp point

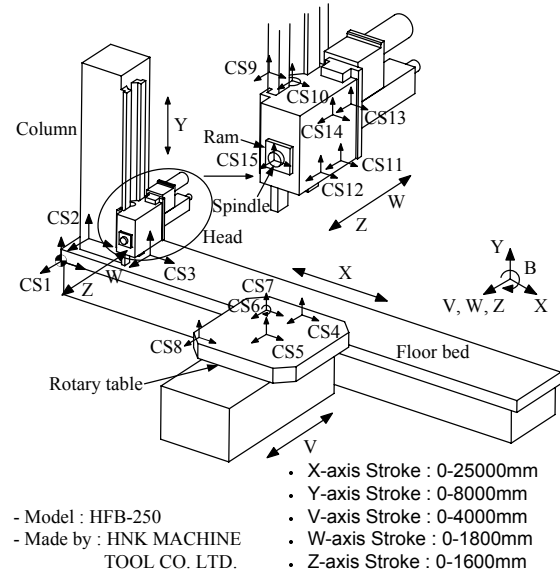


FIGURE 1. Coordinate Systems of the Floor Type Boring Machine.

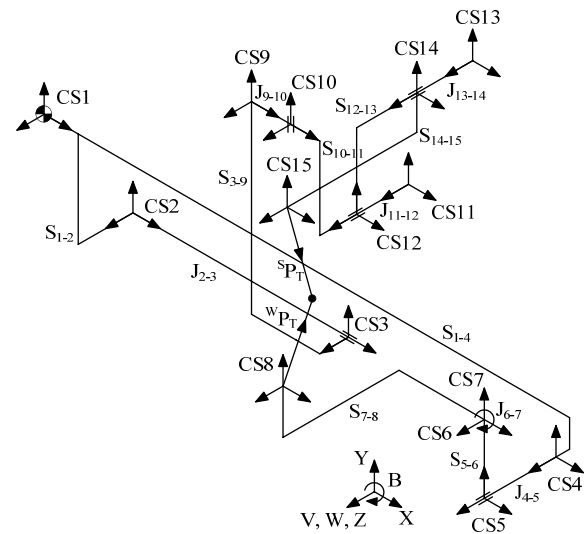


FIGURE 2. Kinematic Chain of the Floor Type Boring Machine.

In Eq. (1), ε_{xi}^s , ε_{yi}^s , ε_{zi}^s ($i=1\sim8$) and ε_{xi} , ε_{yi} , ε_{zi} ($i=1\sim6$) are rotational error terms. Δa_i , Δb_i and Δc_i ($i=1\sim8$) are translational errors due to thermal deformation. S_{xy} , S_{xz} , S_{wy} and S_{zy} are squareness errors. P_x , P_y and P_z are thermal expansions in feed axes. δ_x , δ_y and δ_z are positioning and straightness errors of feed axes. To formulate volumetric errors at cutting point, relative cutting positions with respect to the

machine origin is obtained from two routes as follows:

$${}^R T_T = S_{1-2} J_{2-3} S_{3-9} J_{9-10} S_{10-11} J_{11-12} S_{12-13} J_{13-14} S_{14-15} {}^S P_T \quad (2)$$

$${}^R T_T = S_{1-4} J_{4-5} S_{5-6} J_{6-7} S_{7-8} {}^W P_T \quad (3)$$

In Eqs. (2) and (3), ${}^S P_T = [t_x \ t_y \ t_z \ 1]^T$ is the cutting point with respect to the tool origin, and ${}^W P_T = [X + dX \ Y + dY \ Z + dZ \ 1]^T$ is the cutting point with respect to the work origin. ${}^W P_T$ is derived from the kinematic chain as follows:

$$\begin{aligned} {}^W P_T &= [X + dX \ Y + dY \ Z + dZ \ 1]^T \\ &= [S_{7-8}]^{-1} [J_{6-7}]^{-1} [S_{5-6}]^{-1} [J_{4-5}]^{-1} [S_{1-4}]^{-1} \\ &\quad \times [S_{1-2}]^{-1} [J_{2-3}]^{-1} [S_{3-9}]^{-1} [J_{9-10}]^{-1} [S_{10-11}]^{-1} \\ &\quad \times [J_{11-12}]^{-1} [S_{12-13}]^{-1} [J_{13-14}]^{-1} [S_{14-15}]^{-1} {}^S P_T \end{aligned} \quad (4)$$

dX , dY and dZ are volumetric error terms in X , Y and Z directions, respectively.

TABLE 2. Thermal Gains and Time Constants.

Thermal Constants in Shape Elements			
Gains		Time Constants	
$D_{a_1}^S, D_{b_1}^S, D_{c_1}^S$	1.01, 2.8, 1.6	$\tau_{a_1}^S, \tau_{b_1}^S, \tau_{c_1}^S$	91
$D_{a_2}^S, D_{b_2}^S, D_{c_2}^S$	14.6, 0.8, 0.4	$\tau_{a_2}^S, \tau_{b_2}^S, \tau_{c_2}^S$	47
$D_{a_3}^S, D_{b_3}^S, D_{c_3}^S$	1.3, 5.3, 0.8	$\tau_{a_3}^S, \tau_{b_3}^S, \tau_{c_3}^S$	85
$D_{b_4}^S$	2	$\tau_{b_4}^S$	17
$D_{a_5}^S, D_{b_5}^S, D_{c_5}^S$	2.5, 1.3, 3	$\tau_{a_5}^S, \tau_{b_5}^S, \tau_{c_5}^S$	120
$D_{a_6}^S, D_{b_6}^S, D_{c_6}^S$	0.7, 0.7, 1.4	$\tau_{a_6}^S, \tau_{b_6}^S, \tau_{c_6}^S$	50
$D_{a_7}^S, D_{b_7}^S, D_{c_7}^S$	0.2, 1, 3.6	$\tau_{a_7}^S, \tau_{b_7}^S, \tau_{c_7}^S$	29
$D_{a_8}^S, D_{b_8}^S, D_{c_8}^S$	0.2, 0.3, 5.3	$\tau_{a_8}^S, \tau_{b_8}^S, \tau_{c_8}^S$	21
Thermal Constants in Joint Elements			
Gains		Time Constants	
D_x^J	0.5	τ_x^J	120
D_y^J	1.8	τ_y^J	80
D_v^J	0.8	τ_v^J	100
D_w^J	4	τ_w^J	40
D_z^J	5	τ_z^J	50

SIMULATIONS

Substituting Eq. (1) into Eq. (4), and neglecting rotational error and higher order terms, volumetric errors are derived as follows:

$$\begin{aligned} dX &= \Delta a_1(T) - \Delta a_2(T) + \Delta a_3(T) - \Delta a_5(T) + \Delta a_6(T) + \delta_x(x, T) \\ &\quad + \delta_x(y, T) + \delta_x(z, T) - \delta_x(v, T) + \delta_x(w, T) + P_x(T) \cdot x \\ dY &= \Delta b_1(T) - \Delta b_2(T) + \Delta b_3(T) - \Delta b_4(T) - \Delta b_5(T) + \Delta b_6(T) \\ &\quad + \Delta b_7(T) + \Delta b_8(T) + \delta_y(x, T) + \delta_y(y, T) + \delta_y(z, T) - \delta_y(v, T) \\ &\quad + \delta_y(w, T) + P_y(T) \cdot y \\ dZ &= \Delta c_1(T) - \Delta c_2(T) + \Delta c_3(T) - \Delta c_5(T) + \Delta c_6(T) + \Delta c_7(T) \\ &\quad + \Delta c_8(T) + \delta_z(x, T) + \delta_z(y, T) + \delta_z(z, T) - \delta_z(v, T) + \delta_z(w, T) \\ &\quad + P_z(T) \cdot z - P_z(T) \cdot v + P_z(T) \cdot w \end{aligned} \quad (5)$$

Thermal error elements in Eq. (5) are modeled as first order systems as follows:

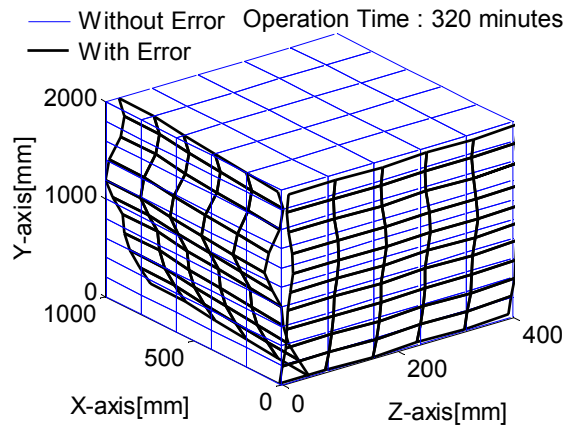
$$\begin{aligned} \Delta a_i(T) &= D_{a_i}^S \left(1 - e^{(-t/\tau_{a_i}^S)} \right) \\ \Delta b_i(T) &= D_{b_i}^S \left(1 - e^{(-t/\tau_{b_i}^S)} \right) \quad \text{for } i=1 \sim 8 \\ \Delta c_i(T) &= D_{c_i}^S \left(1 - e^{(-t/\tau_{c_i}^S)} \right) \\ P_x(T) \cdot x &= D_x^J \left(1 - e^{(-t/\tau_x^J)} \right) \cdot x \\ P_y(T) \cdot y &= D_y^J \left(1 - e^{(-t/\tau_y^J)} \right) \cdot y \\ P_v(T) \cdot v &= D_v^J \left(1 - e^{(-t/\tau_v^J)} \right) \cdot v \\ P_w(T) \cdot w &= D_w^J \left(1 - e^{(-t/\tau_w^J)} \right) \cdot w \\ P_z(T) \cdot z &= D_z^J \left(1 - e^{(-t/\tau_z^J)} \right) \cdot z \end{aligned} \quad (6)$$

Table 2 shows thermal gains and time constants of the shape and joint elements obtained from previous researches [4-5]. Geometric errors of joint elements are constructed based on measurement and previous data [3-4]. Volumetric error analysis is conducted to estimate machining errors in milling and boring processes. Fig. 3 shows machined side surfaces of a rectangular parallelepiped with $(x, y, z) = (1000, 2000, 400)$ mm size. Maximum error is 10 μ m. It is confirmed that machined shapes are affected by thermal deformation for 320 minutes. Fig. 4 shows machined result of a boring operation with diameter 2000mm and length 1500mm for 15 hours. Maximum location errors

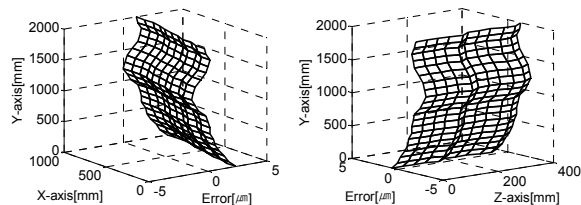
in X, Y, Z directions are $37\mu\text{m}$, $41\mu\text{m}$ and $74\mu\text{m}$. Diametral error for 15 hours is around $30\mu\text{m}$ as shown in Fig. 4. It is confirmed that volumetric errors in boring operation degrade cylindricity a lot.

CONCLUSIONS

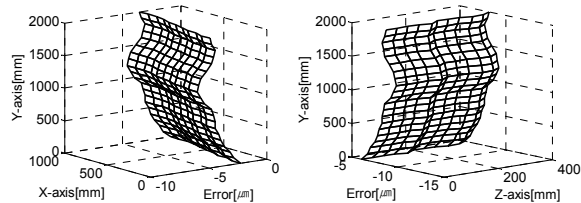
Geometric error model including thermal deformation has been accomplished for a super-size floor type boring machine by using shape and joint transformation matrices. First order responses are applied for modeling the thermal deformations. Volumetric error analysis has been conducted for milling and boring processes. It is confirmed that machining accuracy degrades severely in the super-size floor type boring machine owing to thermal deformation.



(a) Oblique View

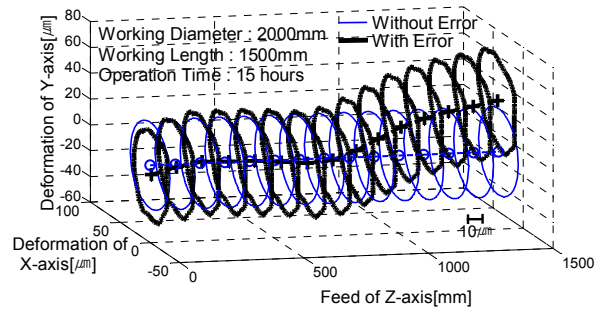


(b) First Machined Surface (c) Second Machined Surface

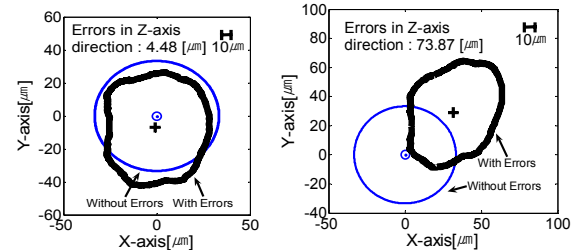


(d) Third Machined Surface (e) Fourth Machined Surface

FIGURE 3. Machining Errors in Milling Operation of a Rectangular Parallelepiped.



(a) Oblique View



(b) Entry Section

(c) Exit Section

FIGURE 4. Machining Errors in Boring Operation of a Cylinder.

ACKNOWLEDGEMENT

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Development of Precision Micromachining Center and Applications Technologies

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INTRODUCTION

In recent years, consumer needs have diversified. The continued pursuit of increased functionality and convenience in optical part, information, communications and medical devices has led to more compact product designs, necessitating the production of smaller and more precise components.

Due to the extremely low productivity and high cost associated with traditional production methods, considerable demand has arisen for new machining methods and machine tools capable of reducing the cost and lead time of precision parts and associated die and mold tooling. In order to meet to this demand, it is necessary to consolidate all machining processes by applying a direct milling approach. To make this a reality, a new machine called "Precision Micromachining Center" (Photo1) was developed by which small diameter cutting tools are used to consistently achieve high efficiency machining of precision parts and tooling components.



Photo1. Precision Micromachining Center (iQ300)

High-Speed/High-Precision feed

The feed mechanism and guide way (feed/guide) system of a precision machine requires extra-fine feed and stability that is greater than a general machining center. Therefore, a feed/guide structure of an ultra precision machine is adopted.

Traditionally, it was possible to satisfy the requirement of high speed and high rigidity by implementing linear guide ways with the use of ball bearings. However, this design introduces vibration caused by the contact of the ball bearing against the guide way surface. To improve this weakness, extra-smooth and high precision roller guides are introduced.

Furthermore, it is imperative to integrate a high-precision profile control function that supports high-speed machining rates while maintaining shape accuracy in sub-micron units.

As an example of machining precision, results of 0.6 μ m roundness (as measured by Taylor Hobson) was achieved for circular machining of a 50mm $\varnothing$ shape with an actual machining feed rate of 2000mm/min. (Fig.1) illustrates the capability of controlling small scale movements where this example shows the result of 0.98 μ m roundness with a 0.1mm radial movement at a feed rate of 200mm/min.

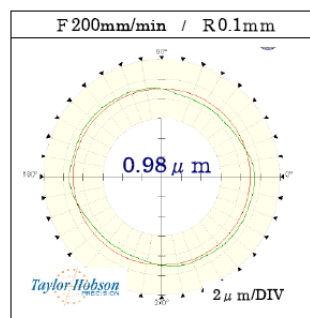


Figure 1. Roundness result of machined circle with radius movement of 0.1mm at 200mm/min

High quality machined surface

In order to achieve optic grade surface roughness with direct milling, it is necessary to minimize the waviness of the guide way structure. The new precision roller bearing/slide way system provides vibration waviness below 30nm. In the case of a directly machined light guide surface (Photo 2), a surface roughness of 5nm Ra (as measured by Zygo NV7300 interferometric profiler) was obtained using a “ruling” style machining process with a non-rotating cutting tool.

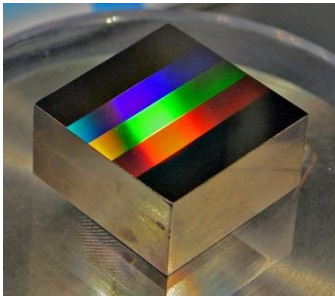


Photo 2. Optical light guide surface produced by “ruled” machining process

In the case of machining with rotating cutters, it is equally important to minimize spindle vibration. In the case of a 45000min^{-1} spindle, with high-accuracy ball bearings, a number of measures are adopted to eliminate unbalance conditions that create vibration.

By adopting additional measures of high precision spindle manufacturing, it is possible to realize significantly improved TIR and NRRO throughout the entire operating range of the spindle. These measures are implemented throughout all steps of the machining, heat treating, grinding, and assembly processes, as well as performing final field balancing of each spindle. The characteristics of these spindles provide the ability to achieve machined surface roughness results of 30nm Ra and below when using cBN ball end mill cutters as shown in (Photo 3).

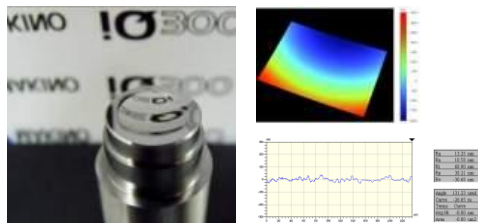


Photo 3. Lens tool insert (54 Hrc) 30nmRa

Long duration machining stability

The linear motor assembly generates heat during operation. In order to neutralize these thermal effects, a double jacket cooling design (Fig.2) was developed and implemented, similar to Makino's patented spindle cooling system. Cooling oil is circulated throughout the linear motor components to control the temperature of the motor so that it matches that of the machine tool. The machine tool is housed in a thermally isolated enclosure to suppress the effects of ambient temperature fluctuation. Additional measures of structural thermal stabilization include glycol circulation throughout the bed and column reservoir as shown in (Fig.3)

To test the effectiveness of these design measures, a rigorous dynamic test was performed over a 12 hour period of time. Measurements confirmed the position of the cutting tool tip was within less than $2\mu\text{m}$ from the original starting position (Fig.4).

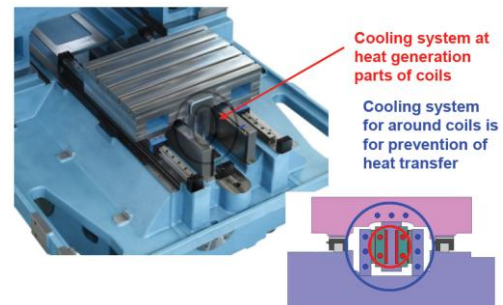


Figure2. Double jacket cooling for linear motor

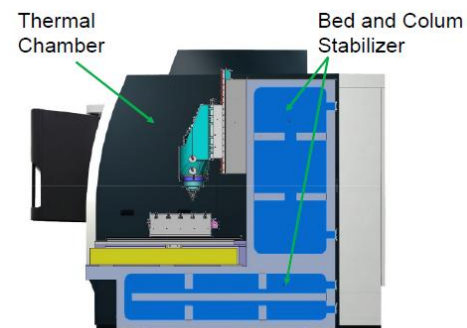


Figure3. Bed and column thermal stabilization

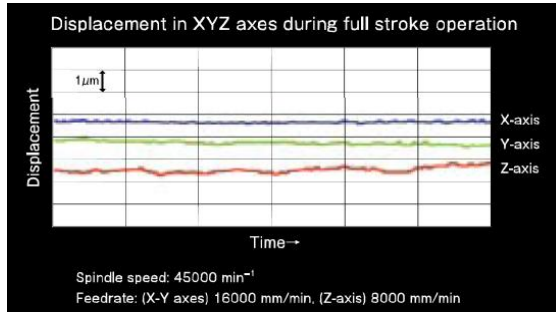


Figure4. Chart recorder results for high speed movement, thermal displacement test

Application of 3D machining for cemented carbide

Cemented carbides possess characteristics of high-hardness, low-thermal expansion, corrosion resistance, abrasion resistance, and provide for the ability to achieve optical grade surface roughness. In the precision micromachining of parts as well as dies and molds, the application of this material has expanded. Typically, these components are produced by applying traditional fabrication processes and manual hand finishing by skilled technicians. A new direct machining approach of utilizing the precision machining center presents opportunities to significantly improve overall manufacturing productivity.

As a testimony to this productivity improvement, a sample die of cemented carbide was produced using the direct milling concept with the precision micromachining center to illustrate the effectiveness of this novel approach (Fig.5).



Figure5. 3D machining of cemented carbide using machining center

Machining application

To achieve a high-precision, high-quality machined surface of cemented carbide, a high-speed spindle with minimized tool vibration and a motion control system that employs high-precision feed with 50nm and lower response capability is required. Additionally, it is important to maintain constant cutting force values through the use of an optimized tool path where cutting volume does not change. In this case, three kinds of diamond tools with unique features are used.

Rough grinding with Electro plated tool

In the rough machining process, a general 3D cavity shape can be generated in a short time, with relatively low cost by utilizing an electro plated grinding tool with high efficiency chip removal characteristics.

Semi-finishing with CVD tool

For the semi-finishing process, it is important to prepare a machining process which leaves a very specific and precise amount of remaining material for the final finishing process. Therefore, the CVD diamond coated end mill with high accuracy tool profile shape is selected.

Final surface finishing with PCD tool

As for the final finish machining process, high quality machining surface and high cutting accuracy will be required. Therefore, the PVD tool with high toughness and abrasion resistance higher than single crystal diamond tool was selected. In preparation for the highly precise finishing process, a special "truing" process is applied to eliminate tool run out and to improve shape accuracy of the cutting tool tip.

(Fig.6) Tool image measuring with CCD sensor) By implementing this tool measurement technology, automatic machining results below 2µm accuracy can be achieved.

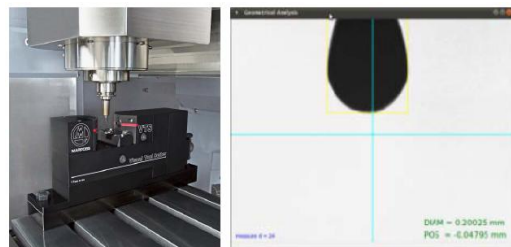


Figure6. Dynamic precision tool image measuring

Comparison of machining time (Fig.7)

The total machining time from the profile forming of roughing process to optical surface of finishing process was 7hours 16minutes on 30mm × 30mm work size. This process provided a reduction of overall machining time of 63% when compared to electric discharge machining.

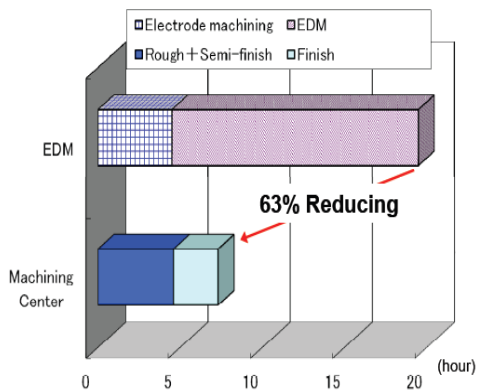
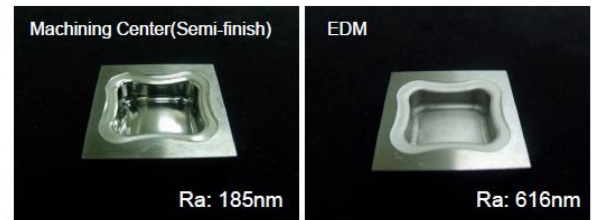


Figure7. Comparison of machining time

Comparison of surface roughness (Fig.9)

The surface roughness of the 3D pocket after finishing process with PCD tool, with special truing of the tool tip was Ra18nm.

For the semi-finishing process, surface roughness was improved by more than 3x when compared with electric discharge machining. In addition, the direct cutting process can create a high quality machined surface without damaged layer by EDM.



a. 3D machining with Milling b. 3D machining with EDMing

Figure9. Comparison of surface roughness

Comparison of manufacturing cost (Fig.8)

By decreasing the machining process and machine tool burden rate of the direct cutting technology, direct machining reduces manufacturing cost of 25%(total manufacturing cost is 3/4) compared with electric discharge machining without hand polishing

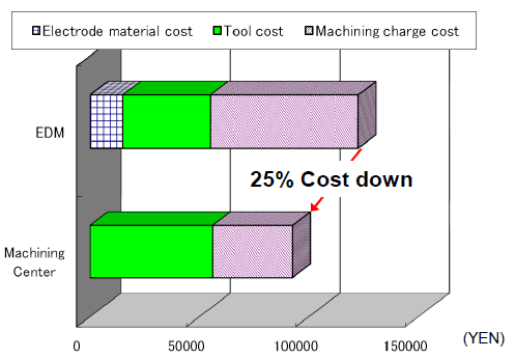


Figure8. Comparison of manufacturing cost

Conclusion

This new machine tool technology dramatically enhances the productivity of manufacturing precision parts, dies and molds as made by the machining center process. In addition to boosting productivity in the field of precision machining technology, these machine tool innovations will facilitate advances in various types of cutting and machining, leading to widespread improvements in other areas.

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CHARACTERIZATION OF A SI-TRACEABLE VACUUM-COMPATIBLE PRECISION NANOINDENTATION PLATFORM

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SUMMARY

This paper discusses the ongoing development of a vacuum-compatible, precision nano-indentation platform (PNP) that is directly traceable to *Système International d'Unités* (SI). In particular, results from recent initiatives to address environmental control, vibration (acoustic and mechanical) and electronic and mechanical stability are presented. Preliminary force-displacement data for Berkovich indentation of silica are also reported.

OVERVIEW OF DEVELOPMENT PROGRAM

Nanoindentation [1] is a leading method for determining a number of mechanical properties from small volumes of materials using indentation depths well below one micrometer. Typical material properties extracted during nanoindentation are: hardness, elastic modulus, work of elastic/plastic deformation, time-dependent material behavior, fracture toughness and residual stress. It can also be used to extract stress-strain curves. Examples of applications include the study of MEMS device components, grain properties in metals, ultrathin coatings and films, and biological materials.

Accurate nanoindentation measurements remain

challenging due to difficulties associated with calibration of both force and displacement and the prediction of uncertainties with current instrument designs. Additionally, the frame, positioning stages and sample-stage interface will all have non-zero compliance that will, in general, depend on force, thereby requiring detailed compensation and further complicating uncertainty estimates. Other uncertainties arise from subsystems, often multi-component assemblies, which will add further compliances in all six degrees of freedom. Finally, each interface between assembled components will exhibit unpredictable instabilities such as creep, differential thermal expansion, different thermal capacities and diffusivities, hysteresis, *etc.*, that in turn depend on the methods of fastening and joining employed.

The goal of this development program is to quantify contributions to the uncertainty by separating and measuring them or, where that is not possible, incorporating conservative estimates. A complimentary goal is to design an instrument that eliminates these sources of uncertainty or has sources of uncertainty that can be reliably predicted or measured through consideration of both the force and displacement measurement paths.

PNP TEST FACILITY

The instrument consists of two major sub-systems; a precision indenter head and coarse translation stages for specimen positioning. To further improve operation and stability of the instrument, a vacuum chamber has been built to provide seismic and acoustic isolation from the laboratory environment as well as thermal isolation to reduce temperature changes during measurement. The vacuum chamber also serves as a partial Faraday cage, further reducing electronic noise and thereby improving measurement resolution.

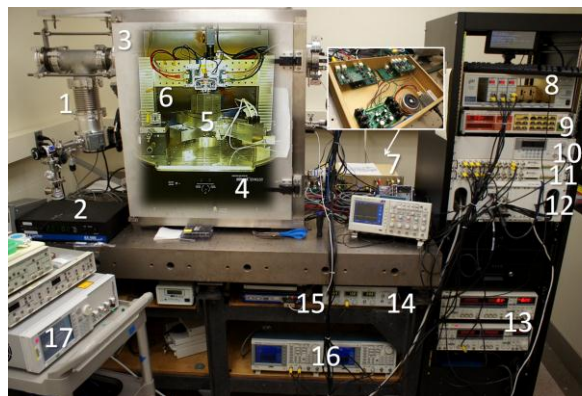


FIGURE 1. Main components of the traceable nanoindentation system.

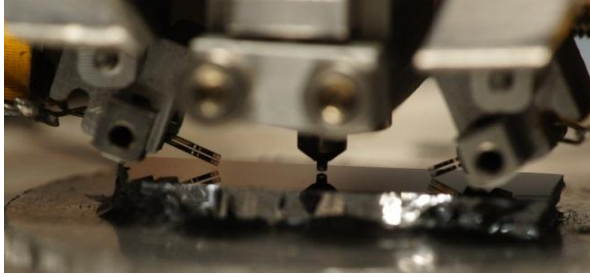


FIGURE 2: Close up photograph of the two tuning-fork tracking probes on either side of a Berkovich indenter tip.

The main components of the system, as shown in Fig. 1, are:

1. Vacuum pump
2. Pressure gauge
3. Vacuum chamber
4. Vibration isolation table
5. Positioning system
6. Frame and indentation head
7. Sensor conditioning electronics
8. Rack with analog filters
9. Precision capacitance bridge
10. Signal generator (drive signal for capacitance gauges)
11. Breakout panel for data acquisition
12. Real-time control system with field-programmable gate arrays
13. Lock-in amplifiers
14. High voltage amplifier for driving surface referencing piezoelectric (PZT) actuators
15. Controller for z-axis nanopositioner
16. Signal generators for tuning fork drive
17. Tunable infrared laser (SI traceable length reference)

This instrument uses a direct-referencing system that transfers the specimen surface location to a common measurement frame that also has calibrated displacement sensors for indenter penetration depth measurement. For a more detailed description of the operation of this instrument see [2]. Using this design, parasitic deformations of the nanoindenter structure, which present the major challenge to uncertainty estimation, have an insignificant effect on the measurement of indenter penetration depth. In addition, the system is capable of achieving sub-nanometer and sub-micronewton force-sensing performance through the incorporation of an ultra-stable laser interferometer system [3], capacitance-based sensing and piezoelectric actuators all controlled using real-time signal processing for data acquisition and analysis.

Surface tracking probe and probe holder

Surface tracking probes and probe holders require a high level of stability, since they are a part of a metrology loop. Therefore, any instabilities will directly couple to the indentation depth measurement. The probes are housed in kinematic mounts with flexure-based alignment capable of positioning the surface probes to better than 2 μm relative to indenter tip plane, as shown in Figure 2.

The surface tracking probes used in this instrument are crystal oscillator tuning forks (TF) with resonant frequencies ranging from 32 kHz to 96 kHz. Each probe is mounted between a V-groove and flat, allowing coarse angular adjustment before it is preloaded with a set screw. To track the specimen surface, oscillators generate sine waves (at frequencies near to individual TF resonance values) that are applied to the TF sensors. The TF response is then amplified and sent to phase detection circuitry. The phase signal is used to dynamically track resonance by changing the oscillator drive frequency. Changes in TF resonance frequency resulting from very weak interactions between the TF and the specimen surface are used by a second feedback loop to control a high-voltage amplifier driving the piezoelectric actuator attached through flexure mechanism to the TF, thereby maintaining the position of these sensors relative to the surface being tracked.

PNP OPERATION

A detailed view of the PNP indenter load head inside the vacuum chamber is shown in Figure 3. A typical indentation sequence can be broken down into stages. In the first stage, a PZT actuator (1) displaces the D-frame (2) of the indenter head towards the specimen (9) until the TF sensors (5) detect the presence of the surface. In the second stage, while the D-frame continues its motion, a pair of PZT actuators (3) expands to keep a constant force between each TF sensor and the specimen surface (as indicated by a constant shift in each resonant frequency). In the third stage, continued motion of the D-frame brings the indenter tip (8) into contact with a specimen surface (9). This contact is detected by the force cell (6). Stage four is the actual indentation process, wherein the D-frame actuator provides the needed displacement and the sensors (5) and servo loops track the surface, compensating for any stage, frame or instrument parasitic deformation. The force is measured by the force cell (6).

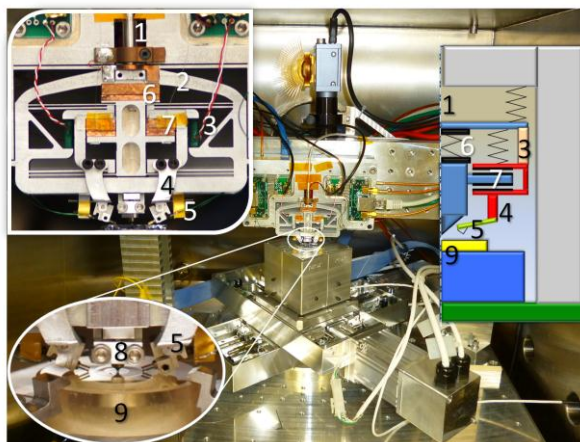


FIGURE 3: Interior of the vacuum chamber. Insets show a close-up of the indentation head (photo and schematic) and the tuning forks and indenter.

Indentation depth is transferred from the TF sensors (5), which sense the surface position next to the indentation region, through a servo control loop to the capacitance gauges (7), which measure a difference between the position of force cell and specimen surface (indentation depth).

ASSESSMENT OF INDENTER ENVIRONMENT

As part of an ongoing initiative to develop traceable indentation instrumentation with nanometer level uncertainties, it is necessary to create a controlled environment in which to operate this facility. Two major environmental contributors to uncertainty are temperature variations and mechanical vibrations (seismic and acoustic).

Vibration Isolation

To achieve nanometer level performance, a significant effort has been invested in vibration attenuation, ultimately resulting in a multi-stage

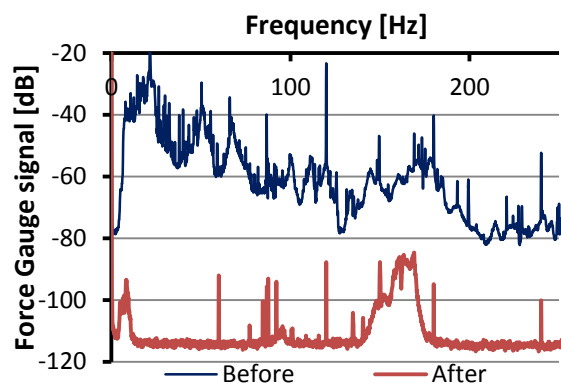


FIGURE 4: Force gauge signal showing frequency spectrum before and after vibration isolation.

isolation system. Figure 4 shows vibration spectra recorded by the force gauge when the system was first assembled in the vacuum chamber (blue) and after adding vibration-isolation components (red). To achieve this level of isolation, passive air isolators were mounted on each table leg and elastomer pads were placed both between the table base and the table top and between the table top and the vacuum chamber. In addition, inside the vacuum chamber the nanoindenter frame was mounted on a passive vibration isolation table. The combined isolation achieved up to -90 dB attenuation at some frequencies.

Temperature Isolation

To reduce the effects of laboratory temperature fluctuations on the stability of the instrumentation (interferometric and capacitance sensors), the PNP is designed to work in vacuum (from 10^{-1} Pa to 10^{-4} Pa).

The vacuum chamber decreases temperature variation by roughly a factor of 100 (see Figure 5), from multiple swings of almost 1 °C in the laboratory to below 0.02 °C measured over a period of 1.5 h. It also enables three modes of instrument operation; in air with the chamber sealed (for low acoustical and thermal disturbance), under vacuum, or under a low-pressure helium atmosphere. The latter mode of operation will help to achieve thermal equilibrium without compromising the stability of either the capacitance or the interferometric sensors.

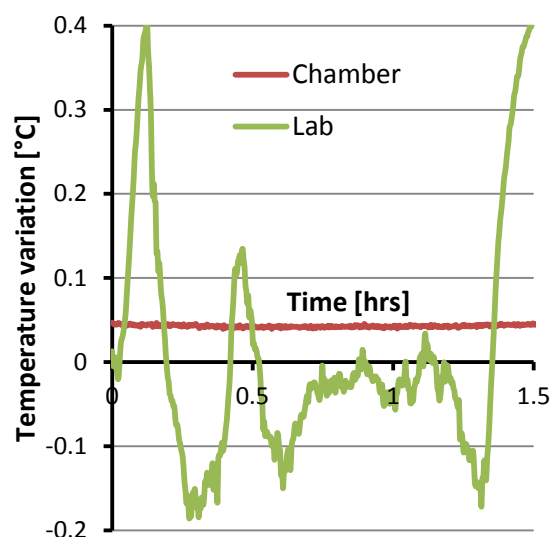


FIGURE 5: Temperature stability of the PNP lab environment and chamber (at ambient pressure).

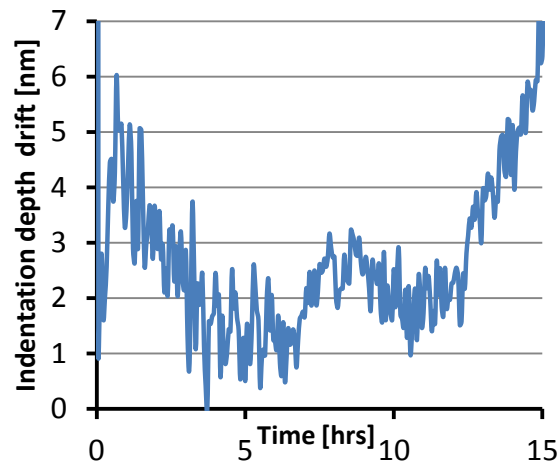


FIGURE 6: Indentation depth drift measurement after 30 mN indentation into Si with force kept constant.

INITIAL RESULTS

To test thermal drifts and effects of the frame compliance in the axis of measurement, a silicon wafer was indented and the force was subsequently held constant at approximately 30 mN for 15 h. Indentation data during this period are shown in Figure 6 for which the maximum recorded change in indentation depth was only 7 nm. During this experiment, the surface tracking system moved more than 150 nm to compensate for drift which, for a typical nanoindenter that does not have surface referencing, would lead to significant error in the indentation depth measurement.

Another experiment was performed to test how well the PNP can repeatedly indent the same material in the same location with increasing maximum force between indentations. The goal was to verify that the unloading slope of force-displacement data (elastic response) collected during first indentation tracks the loading slope of the second indentation. Since work has already been done to deform the material, subsequent indentations will only elastically deform up to the point where the applied force reached the maximum value of the previous indentation. Over this range, unloading and reloading curves should superimpose. Additional plastic deformation will occur only if the maximum applied force is increased beyond level of the previous indentation.

Figure 7 shows six indentations performed in fused silica during which the indenter tip was brought into contact with the material and force

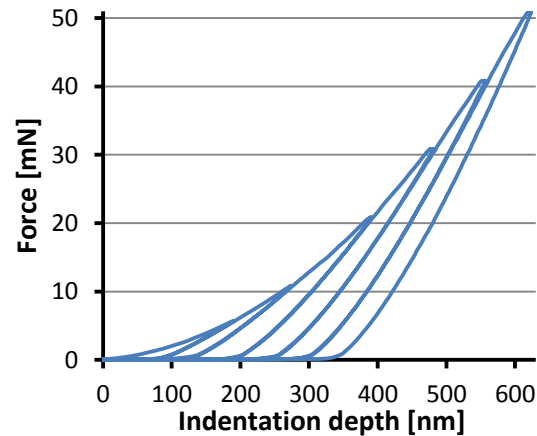


FIGURE 7: Fused silica indentation in the same place multiple times with increasing force between cycles.

was increased until a desired maximum force was reached. After holding this maximum force for a dwell time of 30 s, complete unloading to zero indentation depth occurred. This cycle was repeated with the maximum indentation force being progressively higher for each subsequent nanoindentation. Repeatable force-displacement tracking is observed and is representative of the ability of the instrument to measure material properties and its immunity to instrument compliance and drifts of components outside of the measurement loop. Further work on traceable calibration and determination of combined uncertainties will be reported in future publications.

Disclaimer: Certain commercial materials and products are identified here to specify the experimental procedure. Such identification does not imply endorsement by NIST, or that the material or equipment identified is necessarily the best available for the purpose.

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FABRICATION OF A GREEN CATERPILLAR TYPE IN-PIPE MOBILE ROBOT DRIVEN BY PNEUMATIC BELLOWS ACTUATORS

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INTRDUCTIONS

We have many small pipes whose diameter is about 100 mm. For example, these are gas or water pipes for individual or corporate houses and boilers or hot water pipes for industries. These pipes may be occurred to injure or to corrode by their decrepit. The injury or the corrosion of pipes causes the leak of fluid and links to the accident. So, these pipes must be periodically inspected in order to protect the accident previously. However, they are settled in the ground or in the narrow spaces and some of them are covered by casings or hard heat insulators. Thus, it is very difficult to inspect from outside of the pipe.

If we can insert a mobile inspection robot into the pipe from the suitable position, we are easy to inspect the pipe. In-pipe mobile robots for same diameter pipes driven by the pneumatic actuator have been proposed by the several research groups [1]. However, these pipes are long and their diameters are different at the place where pipes change from the main to the branch. So, the inspection robot for these pipes has to move in different diameters, generate the large traction force for the inspection of the long distance. We have fabricated inspection robots applying an earthworm [2] and an inchworm [3]. The earthworm type inspection robot was designed for the diameter of 79 mm. Then, the robot could not move in pipes whose diameter changes more than 5 mm from 79 mm. On the other hand, the inchworm type inspection robot was confirmed to move in pipes whose diameters are from 70 mm to 140 mm. However, the traction force of the robot changed on the diameters of the pipes. The maximum traction force was 13 N at the 90 mm, 100 mm and 110 mm pipes, but was only 1 N at the 140 mm pipe. The traction force depended on the friction force of the braking mechanism.

So, we propose a suction brake for the braking mechanism. The suction brake can stick on the pipe and hold the pipe by the vacuum pressure. The friction force of the suction brake is larger than those of another braking mechanism and is able to predict easily. Now, we remark the moving motion of a green caterpillar. The green caterpillar sticks a passage such as the leaf by its prolegs. If we can imitate prolegs by the suction brake, we may obtain large traction force. The green caterpillar moves its prolegs by waving its abdomen. This waving motion is imitated by the action of four parallel rubber bellows actuators driven by the pneumatic and the vacuum pressures.

The inspection robot that imitated the green caterpillar can move at the speed of 19 mm/s in the pipe whose diameters are from 70 mm to 150 mm. The traction force of the robot is 14 N and does not depend on the diameters of the pipes. We may be able to inspect long and different diameter pipes, because we have obtained large traction force.

GREEN CATERPILLAR

We observed moving motion of the green caterpillar. The moving motion is shown in Fig. 1. The green caterpillar consists of a head, a thorax and an abdomen. The thorax has six true legs and the abdomen has five pairs of prolegs. Five pairs of prolegs make three groups that are front two pairs, central two pairs and rear one pair of prolegs. The prolegs of the rear one pair are called as claspers. The prolegs are cylindrical. They are used for walking and clinging as they have microscopic hooks on the base.

The moving of the green caterpillar is as follows.

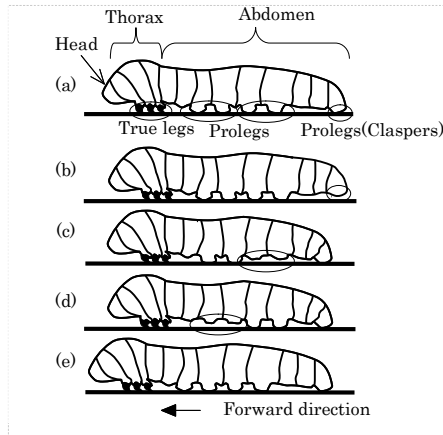


FIGURE 1. Moving motion of a green caterpillar

- In the initial condition, the green caterpillar is sticking to the leaf by all of prolegs of the abdomen and is stationary.
- The rear abdomen waves and the rear one pair of prolegs (claspers) are taken off.
- The central abdomen waves and the central two pairs of prolegs are taken off and the rear one pair of prolegs land on the leaf that is forward direction from the initial condition.
- The front abdomen waves and the front two pairs of prolegs are taken off and the central two pairs of prolegs land on the leaf that is forward direction from the initial condition.
- The green caterpillar stops its waving and sticks the all of prolegs to the leaf. Then, the head and the thorax are moved to forward direction. The cycle of moving of the green caterpillar ends.

We understand that the green caterpillar can move to the forward direction by moving of three groups of prolegs in order to the forward direction.

STRUCTURE OF A ROBOT

The robot moved in pipes is driven by the pneumatic pressures. A structure of the robot is shown in Fig. 2. A fabricated robot is imitated the green caterpillar. The abdomen is imitated by two groups of four parallel rubber bellows actuators which are arranged in the matrix of two rows and two columns. The matrix of two rows and two columns makes a somite of the abdomen. Two parallel rubber bellows in the same row are supplied same pressure by an air feeding tube. The bellows made of Nitrile Butyl Rubber (NBR) is 16 mm in diameter and 55 mm long. A group of prolegs is imitated by a suction

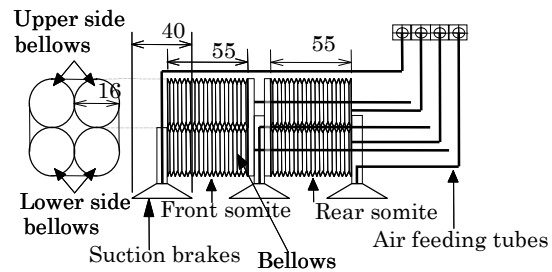


FIGURE 2. Structure of a green caterpillar robot

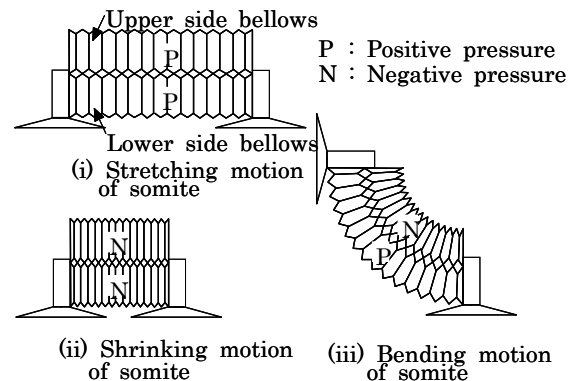


FIGURE 3. Moving motion of somite

brake. The suction brake is a sucker made of NBR and its diameter is 40 mm. The air feeding tube for the positive and the negative pressures is connected in the center of the suction brake. Three suction brakes are arranged at the front, the center and the rear of two groups of four parallel rubber bellows actuators. Total seven air feeding tubes are connected the electromagnetic valves. The electromagnetic valves are attached direct behind of the robot. They are connected two different pressure lines and the signal lines. These lines are carried on conveying mechanisms in order to decrease friction force between the pipe and them.

Moving principle of the somite

Moving principle of the somite is shown in Fig. 3. The somite consists of bellows. The bellows stretches when positive pressure is supplied and shrinks when negative pressure is supplied. That is, the somite stretches when positive

pressure is supplied to the upper and the lower side bellows. The somite shrinks when negative pressure is supplied to the upper and the lower side bellows. Besides, the somite can bend when different pressure is supplied to the upper and the lower side bellows, for example, the somite bends upward, when negative pressure is supplied to the upper and positive pressure is supplied to the lower side bellows. The waving motion of the green caterpillar can be imitated by the stretching, shrinking and bending motion of the somite.

Moving principle of the robot

Moving motions by the pressure changes are shown in Fig. 4. The robot moves to the forward direction by moving its three suction brakes in order to the forward direction. The robot waves its front and rear somite to push the suction brake to the wall of the pipe and to lift up the suction brake from the wall of the pipe. The suction brake can stick the wall of the pipe by supplying the negative pressure and release the wall of the pipe by supplying the positive pressure. In Fig. 4, black colored suction brake shows that the suction brake sticks the wall of the pipe and white colored suction brake shows that the suction brake releases the wall of the pipe.

In the initial condition, the robot shrinks its front and rear somite and sticks its three suction brakes on the wall of the pipe and is in the stationary.

Step 1: The front suction brake releases the wall of the pipe. The front somite stretches to the forward direction. Then, the front suction brake moves to the forward direction.

Step 2: The front suction brake sticks the wall of the pipe.

Step 3: The central suction brake releases the wall of the pipe. In this time, the front somite shrinks and the rear somite stretches. Then, the central suction brake, the electromagnetic valves and the pressure lines move to the forward direction.

Step 4: The central suction brake sticks the wall of the pipe.

Step 5: The rear suction brake releases the wall of the pipe. The rear somite shrinks. Then, the

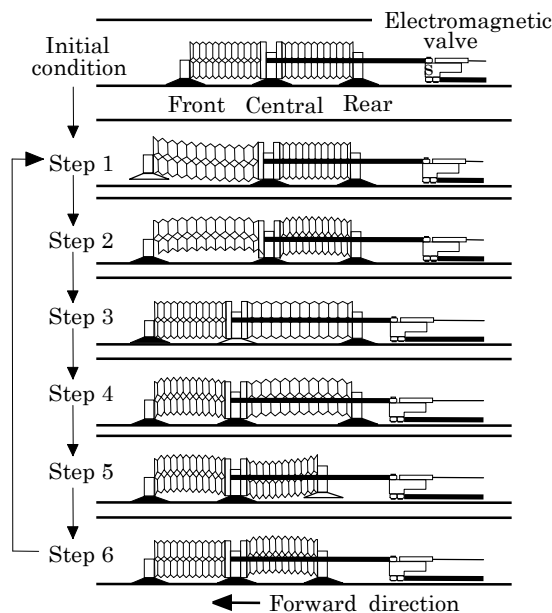


FIGURE 4. Moving principle of robot

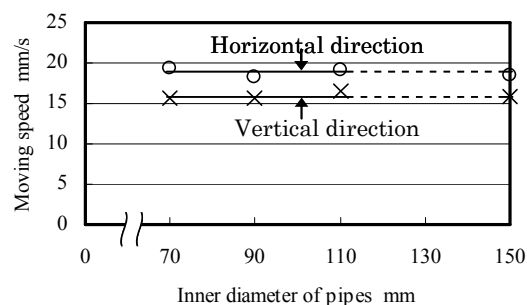


FIGURE 5. Moving speed characteristics

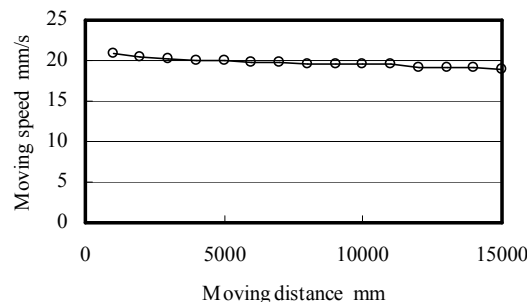


FIGURE 6. Relationship between moving distance and speed

rear suction brake moves to the forward direction.

Step 6: The rear suction brake sticks the wall of the pipe. Thus, the robot moves to the forward direction.

The robot can move in the pipe by repeating the steps.

Moving characteristics of the robot

The positive pressure of 90 kPa and the negative pressure of -84 kPa are used in the experiment and diameters of pipe made of acrylic are within 70 mm and 150 mm.

Moving speed characteristics

Moving speed of the robot is measured in the pipe which is set in horizontal and vertical. The moving speed is shown in Fig. 5. Average speed is 19 mm/s in the horizontal case and 16 mm/s in the vertical case. It is confirmed that the moving speed does not depend on the diameters of pipe by the experiment.

We conducted an experiment in which the robot traveled in a vinyl chloride pipe 79 mm in diameter. The relationship between travel and speed of the robot is shown in Fig. 6. Speed at the start is 21 mm/s, but at 15 m distance decreases to 19 mm/s, so the robot travels 15 m in 13 minutes.

Traction force characteristics

We studied the traction force of the robot, which must move two pressure lines, signal line, and the electromagnetic valves. Traction increases based on the distance traveled. The experimental traction force is shown in Fig. 7. The maximum traction force of 20 N is obtained and does not depend on diameters of the pipe.

Confirmation of passing of the step

The robot is confirmed to pass the step of 5 mm that is made by connecting of pipes whose diameters are 110 mm and 120 mm. The robot can pass the step by its waving motion of the somite

CONCLUSIONS

The green caterpillar can move to the forward direction by moving of three groups of prolegs in order to the forward direction. The green caterpillar moves its prolegs by waving its abdomen.

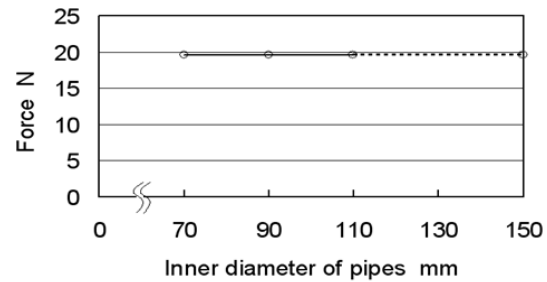


FIGURE 7. Relationship between diameter of pipe and generated traction force

In the fabricated robot, the abdomen of the green caterpillar is imitated by bellows and the prolegs are imitated by the suction brake. The somite structured by four parallel rubber bellows actuators which are arranged in the matrix can imitate the waving motion of the abdomen.

The robot that is imitated the motion of the green caterpillar can move pipes whose diameter are more than 70 mm. Average speed is 20 mm/s in the horizontal case and 23 mm/s in the vertical case. The traction force is 20 N and does not depend on the diameter of the pipe. This type inspection robot may move long distance, because it has larger traction force. The robot is confirmed to pass the step of 5 mm.

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Characterization of Vertical and Pitch Stiffnesses of Porous Vacuum Preloaded Air Bearings

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INTRODUCTION

Demands for high-resolution and high-speed machinery have been pushing the limits of bearing technology. Vacuum preloaded air bearings (VPL's) provide near-frictionless motion and high pitch stiffness against moments that are created when the actuation force or an external work force (e.g., due to cutting), does not align with the center of mass of a moving stage. The VPL uses a combination of air pressure and vacuum, as illustrated in Fig. 1. The vacuum under the bearing structure preloads the bearing against the guide surface, while the pressurized air prevents the bearing from making physical contact with the guide.

While there has been theoretical and practical work done on modeling and characterization of orifice and porous type air bearings [1,2,3,4], porous VPL's have not yet received as much attention in literature. Yet, VPL's are becoming more widespread in the design of precision machines for micro-machining, as well as for other kinds of manufacturing and handling systems. Hence, there is a fundamental need to better understand their operation and their stiffness contribution to the kinematic chain of a machine, which has motivated this study.

The purpose of the research presented in this paper is to collect data and experimentally develop a model to characterize the vertical and pitch stiffnesses of porous VPLs in static loading conditions.

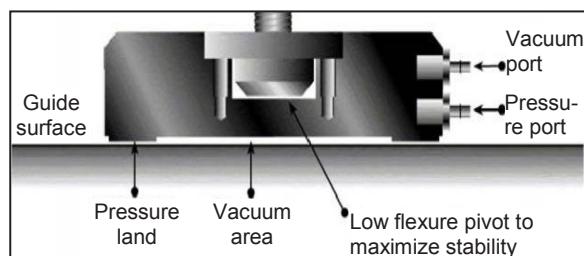


FIGURE 1. Graphical representation of the cross-section of a circular VPL air bearing [5]

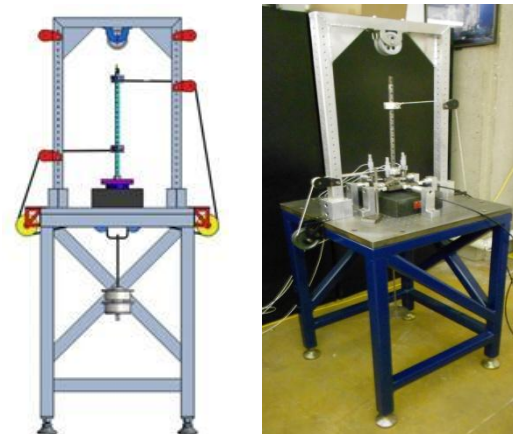


FIGURE 2. Test rig to characterize stiffness in static loading conditions of porous VPLs

EXPERIMENTAL SETUP

A test setup for the characterization of VPLs has been designed and built (Fig. 2) to conduct static loading experiments. Static loads can be applied vertically downwards, vertically upwards, and laterally to generate a torque on the bearing. A configurable pulley system is used to transmit the load from hanging weights to the bearing. The bearing is affixed to a precision-ground steel housing using epoxy and a mast connected to the housing transfers all loads to the bearing. Teflon bumpers keep the bearing in

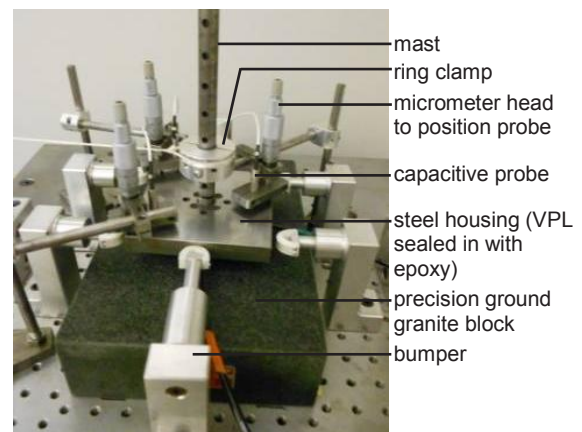


FIGURE 3. Close up view of bearing test area

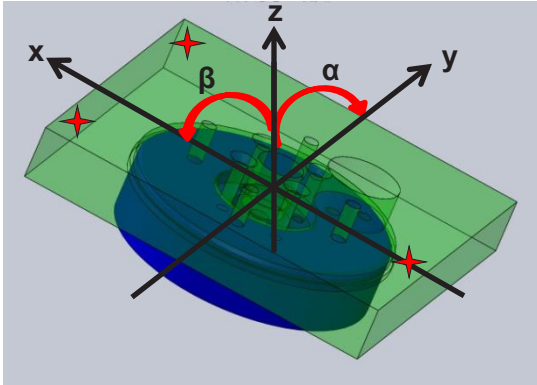


FIGURE 4. Position of capacitive gauges

place as it floats atop a 12"x8"x3" precision ground granite block with 1.2 μm surface flatness and roughness. Care was taken to reduce the effects of friction. Grease was used on the pulleys, and the initially used braided steel rope was replaced with #50 fishing line.

Displacement measurements were obtained using capacitive sensor probes. Lion Precision's CD3 three channel Compact Driver system is used in conjunction with three fine C23-C displacement measuring probes. The sensors provide a measurement resolution of 10 nm RMS with a measurement bandwidth of 15 kHz. The vertical displacement at three points on the steel housing were measured and used to define a plane congruent to the bearing face. Assuming a fixed x, y position for each sensor, the vertical displacements (z_1, z_2, z_3) are measured and the following equations are used to determine the fly height (h), pitch (β), and roll (α) angles:

$$\alpha = \tan^{-1}(b/\Delta), \quad \beta = -a \tan^{-1}(\alpha/\Delta), \quad h = c/\Delta$$

where:

$$\Delta = x_1(y_2 - y_3) + x_2(y_3 - y_1) + x_3(y_1 - y_2)$$

$$a = y_1(z_3 - z_2) + y_2(z_1 - z_3) + y_3(z_2 - z_1)$$

$$b = x_1(z_2 - z_3) + x_2(z_3 - z_1) + x_3(z_1 - z_2)$$

$$c = x_1(y_2 z_3 - y_3 z_2) + x_2(y_3 z_1 - y_1 z_3) + x_3(y_1 z_2 - y_2 z_1)$$

Experiments were conducted on three circular bearings of different diameters made by New Way Precision: S205001 ($\Phi 50$ mm), S207501 ($\Phi 75$ mm), and S209001 ($\Phi 90$ mm). In future work, a nondimensional model will be developed, thus similar shaped bearings are studied. The sensors were calibrated with just the vacuum being applied. The pressurized air was then supplied, and a loading sequence consisting of adding slotted weights ensued. An equivalent unloading sequence followed, allowing observation of any hysteresis effects.

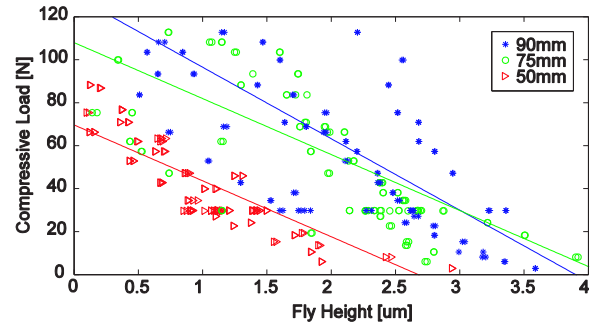


FIGURE 5. Lift-load curve for compressive loading

The experiments were conducted at the rated 60 psi of input pressure and -15 inHg of input vacuum.

EXPERIMENTAL RESULTS

Three main experiments were conducted on each of the three bearings: compressive loading, tensile loading, and applied moment.

Compressive Loading

The loading sequence for these tests consisted of threading the slotted weights around the mast. The results are seen in Fig. 5. Some scatter is observed in the results because in the process of threading the weights onto the mast, the bearing would move slightly. While care was taken to machine the steel housing with high precision, imperfections still exist. Least squares fitting was used to determine the functions in Table 1 which relate fly height, h (in μm), to a given load, F (in N).

Tensile Loading

Application of tensile load was achieved by hanging a weight at the base of the table and using the pulley system to transfer this weight to tensile force pulling up on the mast. The steel housing and mast on the bearing introduced a compressive preload of approximately 30 N. The results are seen in Fig. 6.

A hysteresis effect was observed, most clearly in the $\Phi 90$ mm data. A load force would yield a higher fly height in the unloading sequence than in the loading sequence. This phenomenon was initially attributed to either the air flow patterns beneath the bearing or the mechanical setup.

Table 1. Summary of compression curves

Bearing	
$\Phi 90\text{mm}$	$h = -0.017F + 3.103$
$\Phi 75\text{mm}$	$h = -0.019F + 3.098$
$\Phi 50\text{mm}$	$h = -0.028F + 2.256$

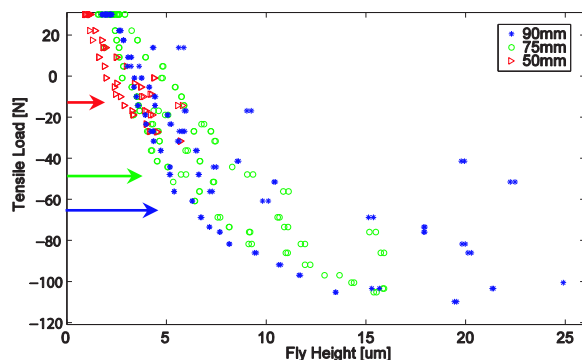


FIGURE 6. Lift-load curve for tensile loading

Gauges and rotameters were inserted in the pneumatic system to record the flow rates of both the pressurized air and the vacuum as the load varied. In a loading/unloading sequence, the flow of the vacuum remained constant while the flow of the pressurized air increased linearly with larger tensile loads. The flow changes showed no sign of hysteresis, indicating that the flow patterns are very likely not the cause of the observed hysteresis.

The teflon bumpers were subsequently studied as potential cause. Each bumper was designed to have two point contacts with the steel housing (Fig. 7), and the material was chosen for its low friction coefficient against solids. A scale was inserted under the bearing and known weights were applied like a compressive test and the output load was recorded. No air or vacuum was applied to eliminate the fluid dynamics factor. The loading force absorbed by the bumpers was found to be 0.94% of the applied force, indicating the bumpers have an insignificant contribution to the hysteresis.

The pulleys also contribute to the overall friction in the system. When the loading force is less than the static friction force F_s , the balance of forces is

$$f_{app} = (k + K)x$$

where k is the stiffness of the bearing and K is the effective pre-sliding stiffness. Once the applied force exceeds F_s , the balance becomes

$$f_{app} = kx + D$$

where D represents the Coulomb friction force.

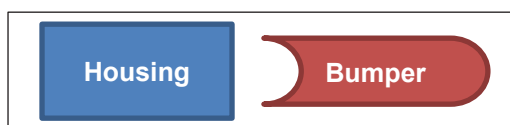


FIGURE 7. Contact between bumper and housing

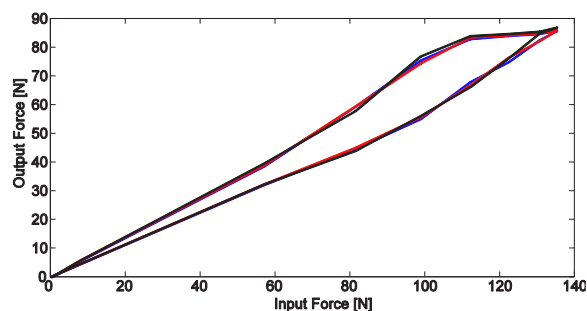


FIGURE 8. Hysteresis evident in friction in pulleys

The overall stiffness for $f_{app} < F_s$ would be $1/(k + K)$, and for $f_{app} > F_s$, the stiffness would be $1/k$. The larger apparent stiffness past F_s would appear as a knee in the lift-load curve.

A similar but reversed effect might be observed in the unloading sequence. Such a knee can be seen in the experimental results for the tensile tests in Fig. 6 (indicated by arrows).

With the bearing still on the scale, an initial compressive preload was applied, and then the sequence of tensile loading and unloading. Again, no pressurized air or vacuum was used in an attempt to isolate air flow related factors. Various loading sequences were used, each repeated three times. Fig. 8 depicts the applied loads versus scale readings. It was found that the curve is highly dependent upon the configuration and current position of the setup.

The results in Fig. 8 definitely indicate there is an inherent hysteresis effect produced by the friction in the pulleys. Due to the high reproducibility of the data in Fig. 8, these values were used to correct the tensile loading measurements. The applied input force was replaced with the actual transferred force, depending on whether the measurement is being taken in the loading or the unloading

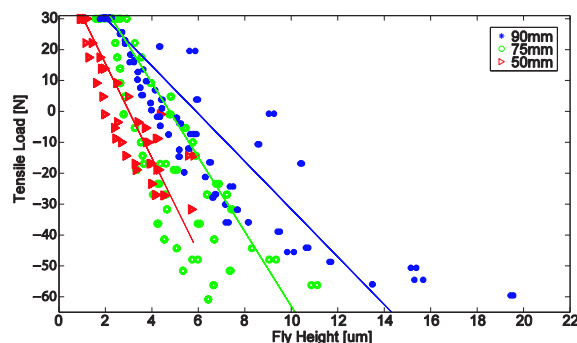


FIGURE 9. Corrected lift-load curves for tensile test

Table 2. Summary of tension curves (corrected)

Bearing	
Φ90mm	$h = -0.1157F + 5.487$
Φ75mm	$h = -0.0744F + 4.502$
Φ50mm	$h = -0.060F + 2.836$

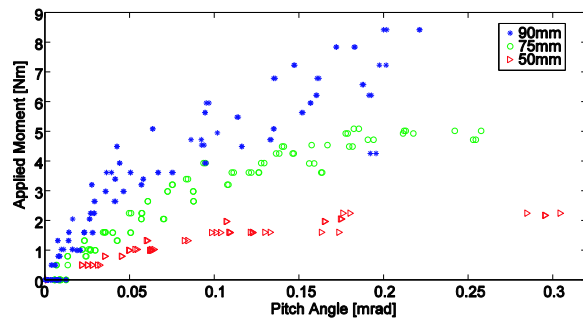


FIGURE 10. Angle-load curve for moment test

sequence. Using this correction, the experimental results from the tensile tests have been redrawn in Fig. 9. The corrected results were analyzed using least squares fitting, constrained to pass through the initial fly height of each bearing. The relationship for the tensile curves are found in Table 2.

Moment Loading

The pitch angle-load curves for the moment test are seen in Fig. 10. The moment was generated around the y-axis by applying a force couple to the mast. The angular deflection about the x-axis remained constant at zero but parasitic angular deflections were recorded. The average parasitic deflection was around 7% of the primary angular deflection. Measurement of the pitch angle around the y-axis demonstrated a quadratic characteristic with applied moment.

A similar hysteresis effect may be observed in the moment results. The correction curve would be different, though, because the moment test configuration does not use the same number or type of pulleys as the tensile test. It is expected that if a similar correction were applied, the relation would also be linear. Currently, additional strain gauges are being added to the setup to measure the actual forces applied to the mast by the different loading lines.

The effective stiffness values for specific ranges, assuming linearity, are reported in Table 4.

Table 4. Summary of stiffness characteristics

Bearing	Φ90mm	Φ75mm	Φ50mm
Tensile [N/um]	7.7	12.0	15.3
Compressive [N/um]	33.3	25.1	26.0
Angular [Nm/mrad] (linear range)	37.5 (0-6Nm)	27.1 (0-5Nm)	14.4 (0-2Nm)

FUTURE WORK

Currently, improvements are under-way to collect higher quality data from the experimental setup in order to help build and experimentally verify a nondimensionalized fluid dynamics model representing the characteristics of the air film. This model will help predict the linear and rotational stiffness of porous VPLs.

ACKNOWLEDGEMENTS

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DEVELOPMENT OF SIMPLE BIAXIAL-TENSILE-TESTING APPARATUS USING LATHE CHUCK

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INTRODUCTION

As the automotive industry addresses environmental concerns, the problem of fuel efficiency and weight reduction has come to the fore. Reducing the weight of automobiles is one of the primary concern by which their fuel efficiency is lowered. The two basic approaches are in automotive design and in materials selection, and they are closely related. Regarding materials, there has been a trend toward the use of not only light metals and also their alloys like as aluminum, magnesium and titanium but also ultra high strength steel in automotive components, particularly automotive bodies. However, such materials in press forming tend to produce fracture, surface deflection, spring back and so on. For this reason, a try and error increases and it has become a cause of a trial production period and cost escalation. The elucidation of the fracture of those materials and the prediction of the forming limit are important technical issues in the field of plastic processing.

The biaxial tensile testing apparatus of the hydraulic control for which the estimation method of material used cruciform specimen is already developed [1]. This apparatus has measured the forming limit and yield loci of various materials with sufficient accuracy. However, in order to pursue performance, apparatus is large-sized and is complicated. Development of a simple and accurate evaluation technique is desired in Small and Medium Enterprise or a manufacturing site. Using the scroll chuck used for a lathe, we developed the apparatus in which tensile with two simple axes is possible and to test. Then, independent chuck by which each axis is independent and operates, arbitrary stress ratio condition are made to act on a specimen, and this study aims at measuring the plastic deformation characteristic with high precision of the apparatus and strain ratio after fraction of sheet metal.

DESIGN OF SPECIMEN

The test material used in this study is an aluminum A1050-O sheet with a thickness of 1mm. Figure 1 shows the geometry of specimen used in this study. The normal strain components, ε_x and ε_y , were measured using scribed circle as shown by Figure 2 [2].

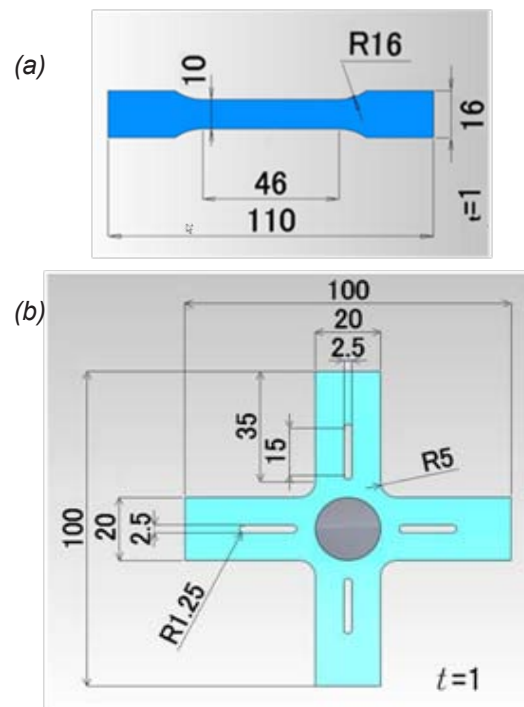


FIGURE 1 Geometry of (a) uniaxial and (b) biaxial tensile specimens

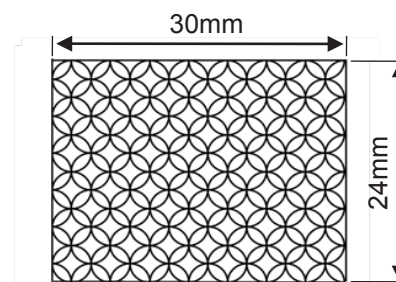


FIGURE 2 Overlapped circles

DATA PROCESSING FOR OVERLAPPED CIRCLES

Figure 2 shows a kind of overlapped circles printed on a specimen, which are blacked in part of in Figure 3 in order to detect those boundaries and intersections by using image sensing camera. The image data taken by the camera are processed to change into binary signals and exclude noises in the image. Two kinds of mask pattern (9X9 pixels) are applied to detect temporarily location of the intersections of each circle.

These intersections are used as the starting and ending points of scanning to find out the boundaries of arcs of the overlapped circles shown in Figure 3; scanning in y-axis direction is carried out to find the transition point of gray level of pixel, which is defined as boundary of a circle. Scanning is moved from intersection A to D, which determines the boundary of the arc AD. This operation is carried out from D to B, A to C, and C to B in a similar manner. These transition points are assembled and approximated by an elliptic equation using the least square method. The intersection of each ellipse a,b,c,d near a temporary intersection, for example A, are obtained as shown Figure 3. Their coordinates are averaged to locate finally the material point e, here final point A. This method leads to have the advantage of the fact that the located point is hardly moved even if the gray level of the circle will changed.

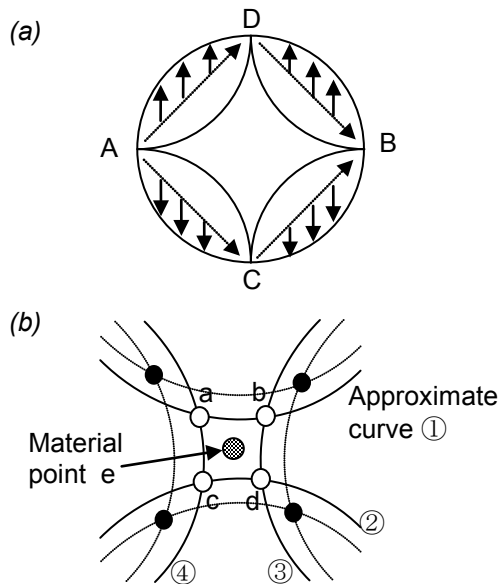


FIGURE 3 Process to detect the boundary of circles (a) and definition of a material point (b).

STRAIN ANALYSIS

It is assumed that a material point is deformed linearly. Then a linear transformation can be applied; the parallel line segments keep straight and parallel to each other even after deformation of the material. Therefore, the ellipse before deformation can be transformed to the ellipse with the different dimensions, where homogeneous strain is obtained. However, the material point on a circle of an ellipse before deformation is not generally on the principal axis of the ellipse after deformation. This fact has not been considered in strain measurement by the conventional scribed circle method, strain of which is estimated by the ratio of the major axes before and after deformation.

The incremental strains can be given by using space coordinates. When the linear transformation is satisfied, two arbitrary straight lines, for example, line segment AB and CD on a circle in Figure 3 can be transferred as shown in Figure 4. Here, the points A, B, C, and D are located at each step of deformation. Component of gage lengths r_{AB} and r_{CD} are as follows,

$$\begin{aligned}\Delta x_{AB} &= x_B - x_A \\ \Delta y_{AB} &= y_B - y_A \\ \Delta x_{CD} &= x_D - x_C \\ \Delta y_{CD} &= y_D - y_C\end{aligned}\quad (1)$$

In the case of points A', B', C' and D' after deformation similar equation are written. Then,

$$\begin{aligned}\alpha &= \tan^{-1} \frac{\Delta y_{AB}}{\Delta x_{AB}} \\ \beta &= \tan^{-1} \frac{\Delta x_{CD}}{\Delta y_{CD}} \\ \alpha' &= \tan^{-1} \frac{\Delta y_{A'B'}}{\Delta x_{A'B'}} \\ \beta' &= \tan^{-1} \frac{\Delta x_{C'D'}}{\Delta y_{C'D'}}\end{aligned}\quad (2)$$

Incremental angle of rigid rotation $\Delta\omega$ is approximated as

$$\Delta\omega \cong \frac{1}{2} \{ \alpha' - \alpha - (\beta - \beta') \}$$

by using

$$\omega = \frac{1}{2} \left(\frac{\partial v}{\partial x} - \frac{\partial u}{\partial y} \right)$$

then $\omega = \Sigma \Delta\omega$

(3)

As gage lengths are changed by deformation, the components are as follows,

$$\begin{aligned}
\Delta x_{A'B'} &= r_{A'B'} \cos(\alpha' - \omega) \\
\Delta x_{C'D'} &= r_{C'D'} \sin(\beta' + \omega) \\
\Delta y_{A'B'} &= r_{A'B'} \sin(\alpha' - \omega) \\
\Delta y_{C'D'} &= r_{C'D'} \cos(\beta' + \omega)
\end{aligned} \quad (4)$$

Where, gage lengths $r_{A'B'}$ and $r_{C'D'}$ obtained as,

$$\begin{aligned}
r_{A'B'} &= \sqrt{(x_B - x_A)^2 + (y_B - y_A)^2} \\
r_{C'D'} &= \sqrt{(x_D - x_C)^2 + (y_D - y_C)^2}
\end{aligned} \quad (5)$$

Therefore, Almansi's strain tensor is obtained as follows,

$$\Delta e_{ij} = \frac{1}{2} \left\{ \frac{\partial \Delta u_j}{\partial x_i} + \frac{\partial \Delta u_i}{\partial x_j} - \frac{\partial \Delta u_k}{\partial x_i} \frac{\partial \Delta u_k}{\partial x_j} \right\} \quad (6)$$

Where,

$$\begin{aligned}
\frac{\partial \Delta u}{\partial x} &\cong \frac{\Delta x_{A'B'} - \Delta x_{AB}}{\Delta x_{AB}} \\
\frac{\partial \Delta u}{\partial y} &\cong \frac{\Delta x_{C'D'} - \Delta x_{CD}}{\Delta y_{CD}} \\
\frac{\partial \Delta v}{\partial x} &\cong \frac{\Delta y_{A'B'} - \Delta y_{AB}}{\Delta x_{AB}} \\
\frac{\partial \Delta v}{\partial y} &\cong \frac{\Delta y_{C'D'} - \Delta y_{CD}}{\Delta y_{CD}}
\end{aligned} \quad (7)$$

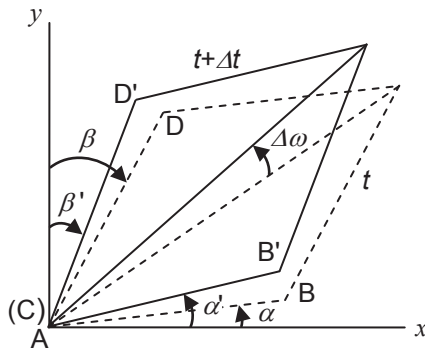


FIGURE 4 Angle of line segment between material points

EXPERIMENTAL PROCEDURE

Figure 5 s shown the experimental setup which consists of three portions. They are Independent chuck, motors, and vice1 - vice4 chucking the specimen. The vices are moved by the motor controlled by the personal computer. The normal tensile speed is 5mm/min.

First, uniaxial specimen shown in the figure 1 was attached to the vice1 and vice3, test was done in consideration of the rolling direction of aluminum. Next, biaxial specimen is attached to

the all vices. Each cruciform specimen was subjected to proportional biaxial strains with $\epsilon_x : \epsilon_y = 1:1, 2:1, 3:1$ and $3:2$, to determine the forming limit diagram.

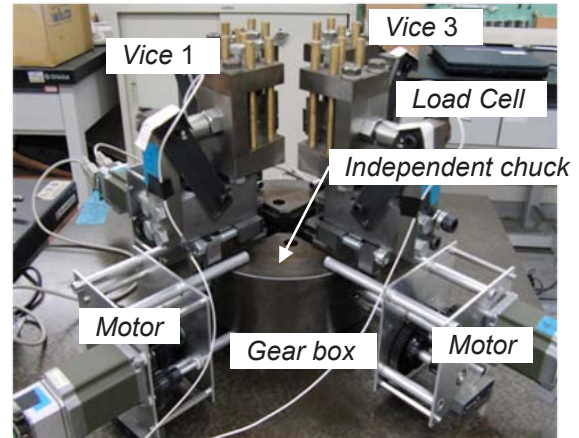


FIGURE 5 Simple biaxial tensile testing apparatus set on the independent chuck

RESULTS AND DISCUSSION

Figure 6 shows the specimen before and after deformation. The stress-strain curve is shown in Figure 7. It was shown that experimental equipment has the performance as well as the usual tensile testing machine.

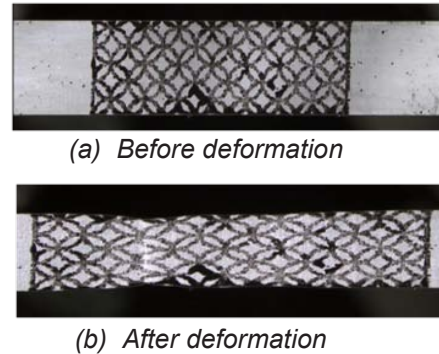


FIGURE 6 Results of uniaxial tensile specimens

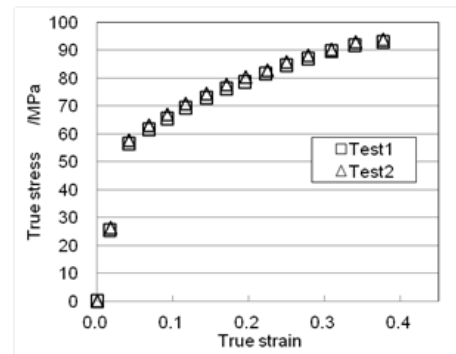
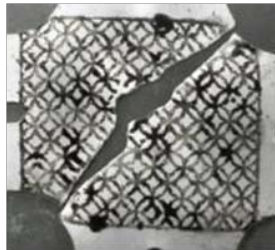


FIGURE 7 Stress-strain curve of uniaxial tensile testing

Figure 8 shows the cruciform specimen before and after deformation. Figure 9 shows the load – major strain curve in measuring time. A figure shows that the measuring loads 1-3 and 2-4 are almost equal. The forming limit diagram is shown in Figure 10. As compared with a general fracture strain, it became a small. Two reasons can be considered to this result. First, the scribed circle's diameter was used by 4mm, then the fracture strain was estimated small since the strain gradient is large near the point of the fracture. Next the hollow was processed on the central part of the cruciform specimen. Thickness of the hollow portion was not uniform strain distribution has occurred.



(a) Before fracture



(b) After fracture

FIGURE 8 Situation of the specimen by biaxial tensile testing

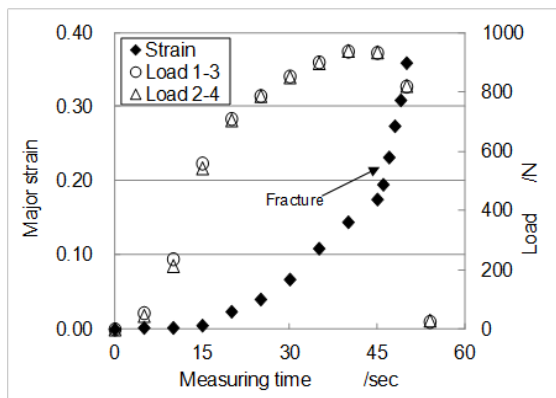


FIGURE 9 Load – major strain curve in measuring time (strain ratio is 1:1)

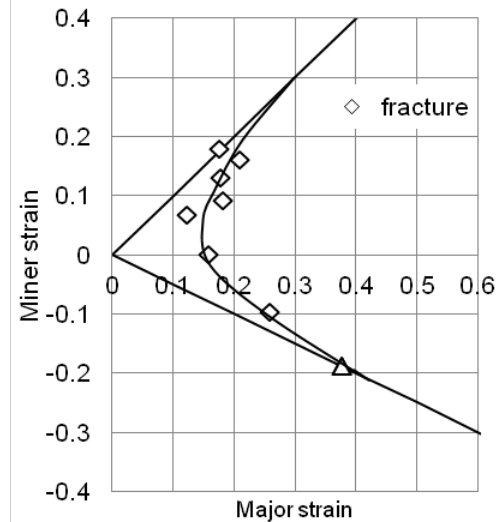


FIGURE 10 Forming limit diagram

CONCLUSIONS

The uniaxial and the biaxial tensile test of A1050-O sheet metal were done using the simple biaxial – tensile– testing apparatus which operates independent lathe chuck. Stress-Strain curve is obtained as good as the conventional uniaxial tensile testing machine. On the other hand, about biaxial test, it became clear that the shape of a specimen needs to be improved. However, arbitrary stress courses were made to act on a specimen, and it became possible to measure the plastic deformation characteristic with high precision.

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Kinematic Solar Dish – Initial Results

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Abstract

This paper is a follow up report for the paper submitted to the 2011 ASPE annual meeting describing a unique kinematic solar concentrator dish design.

The paper documents initial results from a 5 m reflector, and details several measurement techniques designed to provide a complete geometrical characterization of the structure quickly, cheaply, and under field conditions.

The paper also discusses plans for a 12 m parabolic reflector, and how measurement instruments are modified to cope with the larger diameter.

Alignable Carrier Structure

Heliocentric's replaces the traditional static truss and alignable reflector facets system with a tensile kinematic adjustable structure (A bicycle wheel), and snap-in reflector facets. The structure is referred to as the Alignable Carrier Structure (ACS)

Since alignment is achieved at the truss level, the accuracy can be characterized even before the reflective facets are installed onto the ACS.

[CAD image of ACS]

ACS Construction

The ACS is assembled in place, from components that can be conveniently handled by a single person. Typical assembly time for a team of two people is 30 minutes.

Since the structure is kinematic, there is no dependence on part tolerances in order to guarantee that mating features match up.

[Photographs of the construction process]

ACS Measurement and Alignment

Once assembly is complete, we perform the alignment process, which closely mimics the truing of a bicycle wheel – adjusting the length of

the tensile spokes in order to make the rim of the structure flat and circular.

In order to establish geometrical reference in space, two intersecting laser beams exit a tube which lies on the axis of the ACS. These beams intersect at a point that is in the nominal plane of the rim and a known distance from its circumference. As the tube rotates, the beams describe a circle surrounding the nominal position of the rim.

The two beams are intercepted by a dual linear-CCD sensor unit which is mounted to the rim points. By subtracting and averaging the two readings, the unit tells us whether the rim point is in plane or not, and whether its distance from the center axis is correct.

[Images of ACS measurement setup and alignment process]

Results

Flatness and circularity of 5 m reflector, accuracy of measurement.

Next step

Plans to improve measurement method, plans to transition to 12 m reflector.

References

- [1] Shelef B, Erez S, *Precision Engineering In Concentrating Solar Systems*, ASPE annual meeting 2011

NON-CONTACT POSITION CONTROL VIA FLUID SHEAR FORCE

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INTRODUCTION

This paper describes the design and testing of a device that can control the position of a flat wafer or glass round without any contact. The flat object is placed between two horizontal flat air bearings with a small gap (<100 microns) from bearing to object. The top air bearing is actuated using piezo actuators about θ_x and θ_z where the y-axis is the vertical axis. Figure 1 is a photo of the device that was built and tested.

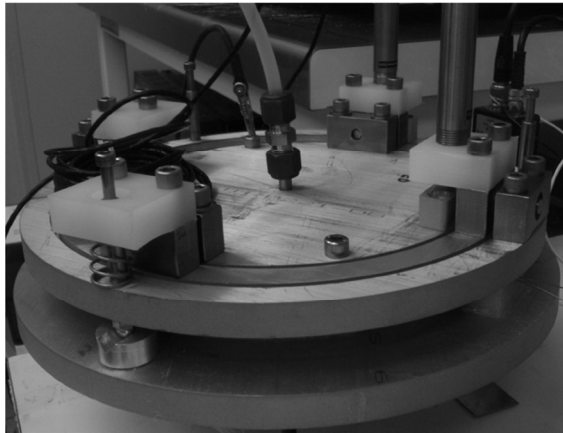


FIGURE 1. Photo of device used to control glass position between two air bearings.

A low pressure (<1 psi) is used to avoid significant deflection of the air bearings. In addition to providing frictionless lubrication, the fluid flow also results in a shear force on the surfaces of the flat object. This shear force is used to control the position of the flat object. By changing the angle of the top bearing with respect to the bottom bearing, the magnitude and overall direction of the shear force can be affected. This is because the side with the smaller bearing gap will have a higher pressure

than the side with the larger bearing gap. The greater the angle, the greater the shear force.

To sense the position of the flat object, an optical system is used in which light is emitted from a fiber directed normal to the edge of the flat object is reflected by the edge, returning into a second fiber juxtaposed to the first and is detected by a photodiode.

MOTIVATION FOR DEVELOPMENT

The intended application for this positioning technology is for non-contact slumping, or shaping, in an oven at 600°C, of glass sheets that are to be used for x-ray telescope mirrors. Traditional contact-slumping methods run into the problems of dust particles settling in between the mandrel and the glass sheet, resulting in local deformities. A non-contact slumping method, such as the two air bearing system detailed above, does not have the issue with particles. However, some form of control is required to keep the glass sheet from sliding out of the bearings, as the system is inherently unstable.

Initially, a simple solid retainer that would physically confine the glass sheet was attempted. However, this resulted in the edges of the glass sheet being pressed against one or two edges of the retainer, which resulted in buckling of the glass sheet that was visible under light as continuous ripples. Higher pressures were also attempted in an effort to force the glass sheet to stay flat, but this instead resulted in larger ripples. From these trials, it was concluded that any form of normal force against the edge of the glass sheet would result in unwanted deformations. Using fluid shear force circumvents this issue.

It should be noted that for the final high temperature slumping device, all components used with this prototype will be replaced with analogous high-temperature components.

Other applications of this technology could include frictionless positioning systems of any objects with flat, cylindrical, or spherical surfaces, or even non-contact processing or cleaning of substrates using high pressured fluids [1].

MECHANICAL COMPONENTS

The device can be divided into two subassemblies, top and bottom, which are mated together through an adjustable kinematic coupling (KC). Each subassembly contains a porous bearing and plenum chamber. Three capacitance sensors provide bearing-to-bearing gap distances. The top subassembly consists of bearing/plenum and two outer rings, each of which provides rotation about an axis. That is, the outer ring contains the upper half of the kinematic coupling and stays fixed to the bottom subassembly. The inner ring is mounted to the outer ring via two shafts and needle bearings, and rotates around θ_x . Then finally, the bearing/plenum is mounted to the inner ring similarly and rotates around θ_z . Two piezo actuators mounted on the inner ring push on the outer ring and the bearing/plenum separately to provide rotational movement. Figure 2 illustrates exaggerated kinematics of the top subassembly, while figure 3 shows a cross-sectional view of the complete assembly.

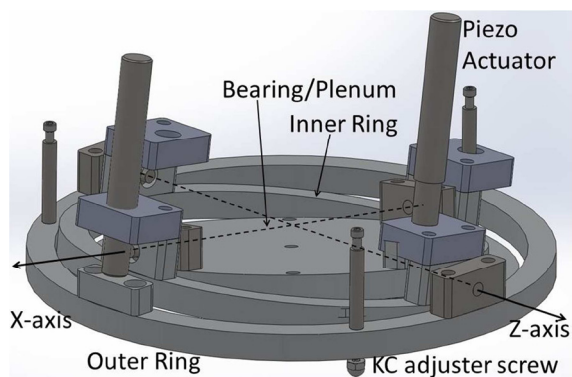


FIGURE 2. Top subassembly of device with exaggerated movements.

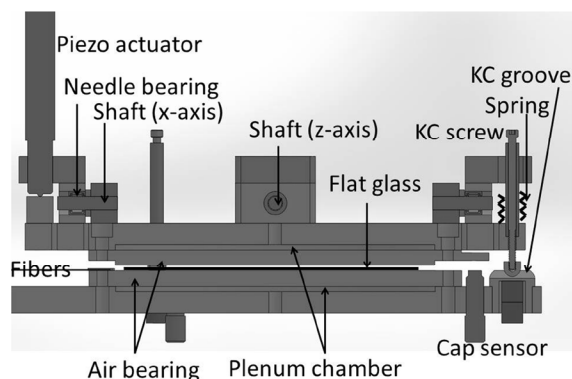


FIGURE 3. Cross-sectional view of the device.

OPTICAL COMPONENTS

A 30 mW 850 nm diode laser is used in the optical system. Using a fiber splitter, the light is split into two beams, which then couple into 0.4 mm-diameter sapphire fibers. These sapphire fibers were chosen because of their ability to withstand high temperatures, specifically up to 600°C. The fibers have a copper coating that oxidizes at 600°C, but our heat testing of the fibers indicates that the sapphire core itself survives the process. Additionally, isolation of the fiber from atmospheric oxygen will increase longevity.

The sapphire fibers are directed perpendicularly to the edge of the glass round and to each other, providing sensing in the x and z axes. The output light bounces off the edge of the glass and is received by a second sapphire fiber juxtaposed to the first fiber. An amplified photodiode at the end of each second sapphire fiber provides a nonlinear voltage that can be calibrated to indicate distance. Figure 4 depicts the optical setup.

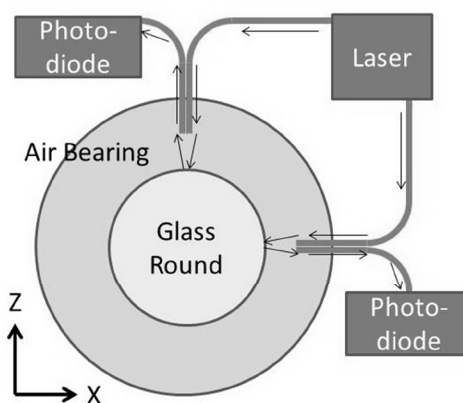


Figure 4. Diagram of optical system in device. Mechanical components not shown.

$$x = 0.8 \cdot V^{0.7} \quad (2)$$

There are two limitations to the current optical setup. The first is that because we are using two separate fibers to input and output light, the light coming into the second fiber reflects off the edge of the glass at a slight angle, instead of at 90° . Therefore the light output starts to decrease when the edge of the glass moves too close to the fiber pair. This begins to occur at about 2 mm away from the fiber pair.

The second limitation is due to the round shape of the glass. Because the edge of the glass is not straight, the distance data of each axis is parasitic. That is, the movement of the glass in one axis affects the light output in the other axis. For the best accuracy, a tedious calibration method involving the recording of both axes' light outputs at points in a point grid must be utilized. In addition, at significant displacements (>10 mm) away from the fibers in one axis, the light reflects at a significantly large angle in the other axis, causing failure of position control due to insufficient light output. Therefore, position control is limited to a certain range. This limit has been ascertained to be about 10 mm away from each fiber pair. For the purposes of this experiment, it has been assumed that the effect of translation in one axis on the other can be ignored for the small range of displacement of glass that is being actuated. Although this affects the accuracy of the positioning system, it does not affect the precision.

CALIBRATION

Using an earlier 1-dimensional test prototype in which the glass was allowed to translate in only one axis, the photodiode voltage was recorded in increments of distance (from the edge of the glass to the fibers). This calibration was performed with a straight edge. As the plot in figure 5 shows, the relationship is nonlinear and is well fit to a power function given by the following equation

$$x = A \cdot 5.56 \cdot V^{0.56 \cdot B}, \quad (1)$$

where V is the photodiode voltage and x is the distance from the glass to the fibers. A and B are calibration factors for the power of the laser and straightness of the glass edge, respectively. Although it was decided to ignore the dependency of one axis on the other, it was found that adding a small calibration factor for roundness of the glass edge helped when implementing the controls system. With calibration factors attained experimentally, the relationship used in the controls system is

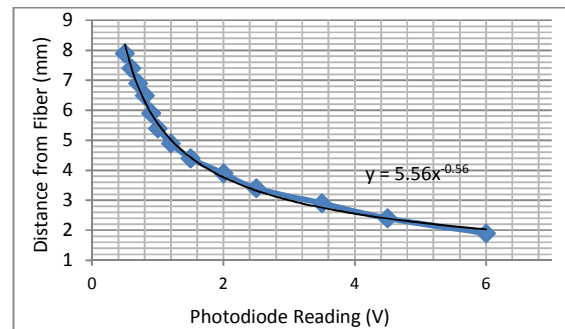


FIGURE 5. Photodiode voltage vs distance from fiber calibration curve.

FREQUENCY RESPONSE

System identification was done on a single axis with a 3 micron-amplitude sinusoidal signal applied to the piezo actuator, while the other axis was undisturbed. The measured transfer function is the ratio of the total signal (sinusoid plus PID compensation) to the piezo, which is directly related to the angle between the two air bearings, to the distance from the glass edge to the fibers. Figure 5 is a Bode plot of the plant transfer function after the PID controller was divided out from the measured transfer function. Due to inefficient Labview code, the program loop could only be run as fast as 5 Hz, inducing a time delay. The Bode plots appear to indicate two poles at about 0.15 Hz.

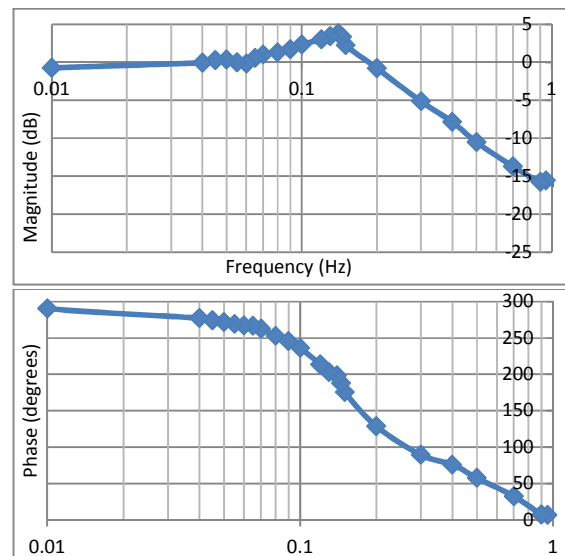


FIGURE 6. Magnitude (top) and phase (bottom) plots of plant transfer function.

CONCLUSION

In this paper, it has been shown that shear force can be used to control the position of a flat object. In addition to applying shear force on the glass surface, the angle of the air bearings also appears to be correlated to a spring and damper action.

FUTURE WORK

One way that the optical distance sensing system might be improved is to double the number of fiber pairs. Two pairs would be used per axis, and would be placed at 90° intervals around the glass sheet. Averaging the photodiode voltages from two opposite facing pairs of fibers would provide a much more accurate reading and could also possibly eliminate the previously mentioned parasitic error.

REFERENCES

[1] Devitt AJ, inventor; New Way Machine Components Inc., assignee. Method and apparatus for in-line processing and immediately sequential or simultaneous processing of flat and flexible substrates through viscous shear in thin cross section gaps for the manufacture of micro-electronic circuits or displays. United States patent 8123868. 2012 Feb 28.

SYSTEM ENGINEERING TO MEET CONTRADICTING REQUIREMENTS

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INTRODUCTION

High accuracy, high thermal loads and high vacuum environment seem to be contradicting requirements for the design of a system. By separating functions we found solutions to meet these requirements. This approach requires “systems thinking” on system level. VDL ETG and Settels Savenije van Amelsvoort designed and built a vacuum system ($D \times h = 1.5\text{m} \times 1\text{m}$) for a light source for EUV lithography, where mechanical tolerances are within $100\text{ }\mu\text{m}$ absolute and less than $10\text{ }\mu\text{m}$ on local level.

FUNCTIONS

Figure 1 shows the working principle of a Laser-Produced Plasma EUV source.

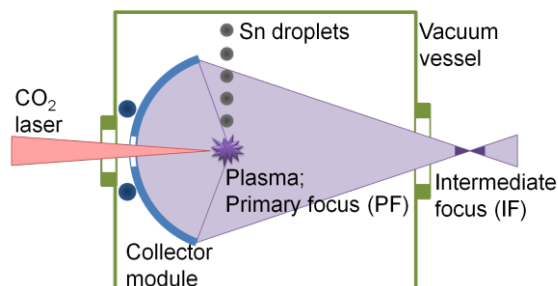


FIGURE 1. Working principle of a Laser-Produced Plasma source

A CO₂ laser beam hits Tin droplets in the Primary Focus of the collector module and an EUV light emitting plasma is produced. The emitted EUV light is focused in the Intermediate Focus point by a Collector Module and delivered to the scanner. The functions of the EUV source that can be distinguished are:

1. Creating plasma
2. Focus emitted EUV light in Intermediate Focus
3. Create optical axis; an accurate and mechanical interface to Intermediate Focus

4. Create vacuum environment, with extreme low Oxygen and Water vapor pressure level
5. Temperature control; absorb heat generated by the plasma

VACUUM VESSEL AND METROLOGY FRAME

The power of the CO₂ laser is in the order of tens of kilowatts, the power of the produced EUV beam is less than 500W. Almost all the power of the CO₂ laser should be transferred through the vessel wall and absorbed by the cooling water. The thermal expansion of the vacuum frame as result of the heat load from the plasma is in the order of 0.5 mm. Also deformations caused by vacuum loads are too big to meet the accuracy requirements.

From this it can be concluded we need to separate functions. A stiff metrology frame is introduced to create an optical axis and inside this metrology frame the water-cooled vacuum vessel is mounted to create a vacuum environment. In figure 2 an overview of the system is shown.

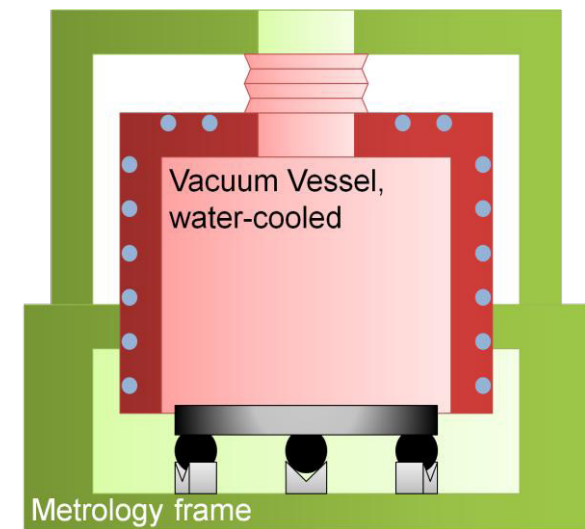


FIGURE 2. System overview

Heat load and vacuum forces are absorbed by the vacuum vessel; accuracy is determined by the metrology frames. These unloaded frames are on room temperature while the vacuum vessel contains high thermal gradients.

COLLECTOR MODULE INTERFACE

Part of the tolerance stack is the Collector Module interface. The Collector Module is mounted in the metrology frame. The requirements on absolute accuracy are high, but also the requirement on repeatability is strict when replacing the component. A stiff kinematic mount is designed: 3 balls in 3 V-grooves, with the thermal centre (TC) in the focus of the Collector Module.

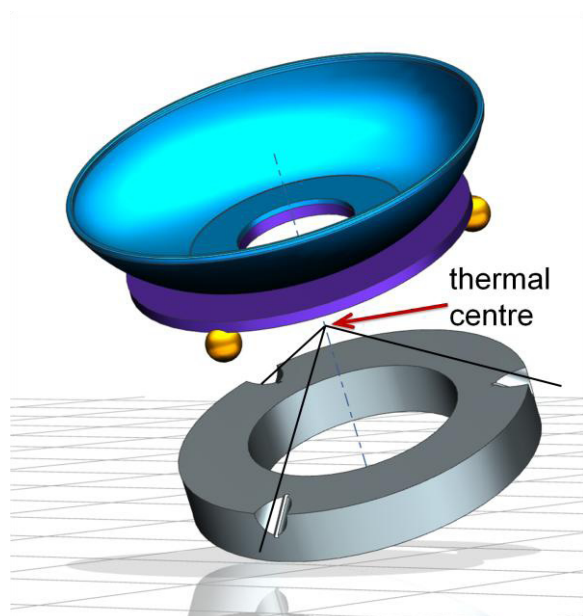


FIGURE 3. Kinematic mount with thermal centre (exploded view)

Contact stiffness

Si_3N_4 balls on hardened tool steel V-grooves (AISI 420) give a Hertzian contact stiffness of $3 \cdot 10^8 \text{ N/m}$ [1]. The orientation of the V-grooves is so that the surfaces of the V-groove form 3 perpendicular surfaces. Now the translational Hertzian stiffness is approximately equal in 3 degrees of freedom; two times the contact stiffness in a contact point: $6 \cdot 10^8 \text{ N/m}$.

Coefficient of friction

Self-alignment is the situation that the Collector Module is aligned by gravity forces. The allowed

coefficient of friction for self-alignment is evaluated for the V-groove configuration and is about 0.4 [-]. Measurements are done on the coefficient of friction of Si_3N_4 balls on Diamond Like Carbon (DLC) coated hardened tool steel and the results are shown in the graphs in figure 4. The measured coefficient of friction is factor of 10 lower than the maximum allowed coefficient of friction for self-alignment.

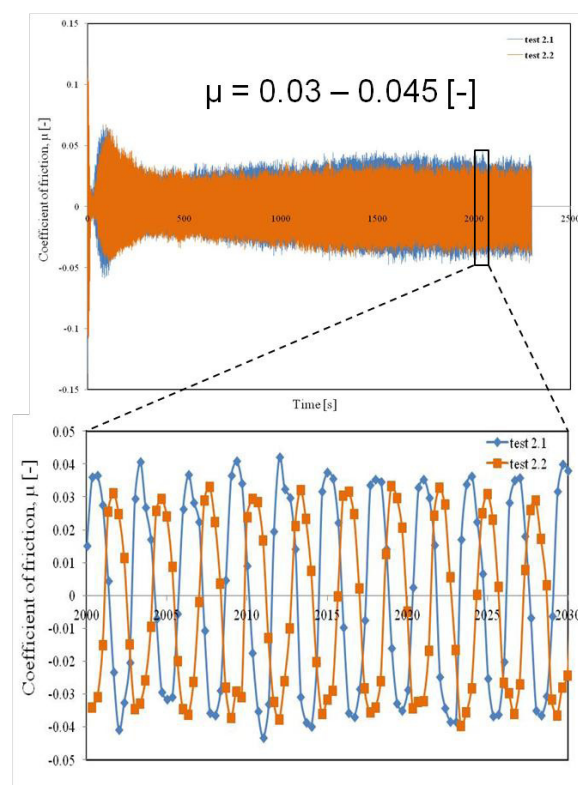


FIGURE 4. Results of oscillating friction test in vacuum environment [2].

Repeatability measurements

Repeatability measurements of the V-grooves interface are done. A dummy Collector Module with three reference balls is placed on the V-grooves six times and the locations of the reference balls is measured on a coordinate measurement machine in x,y and z. The deviation from the average values of the locations of the reference balls are plotted in the diagrams for reference ball 1, 2 and 3. See figure 5. The repeatability of the collector mount is better than 7.5 micrometer.

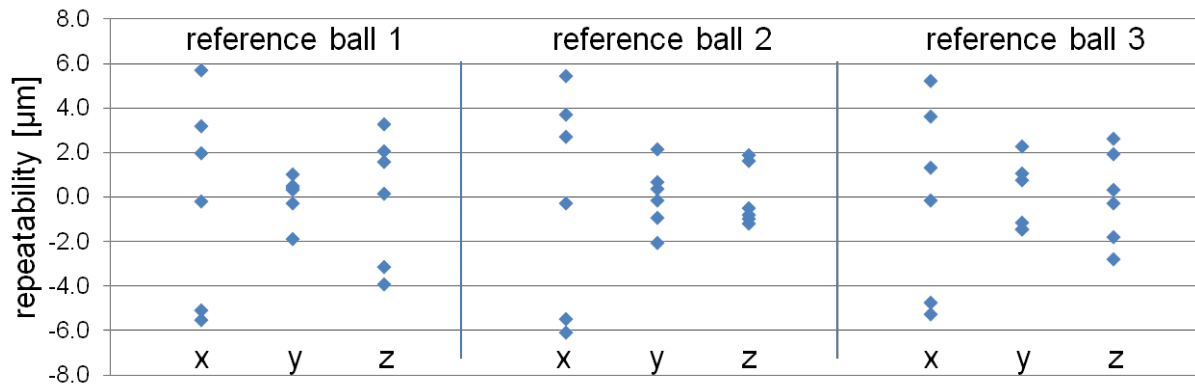


FIGURE 5. Results of repeatability measurements

OPTICAL AXIS MEASUREMENTS

After assembly the error of the interface to Intermediate Focus is measured. Therefore the Collector Module interface and the interface to Intermediate Focus are measured with a coordinate measurement machine (CMM) and a perfect Collector Module is assumed to transfer the measured values of the Collector Module interface to the Intermediate Focus interface. The error of the interface to Intermediate Focus with respect to the optical axis of the EUV beam is less than 100 μm radial.

CONCLUSIONS

By separating functions it is possible to meet specifications that seem to be contradicting.

With a vacuum frame that absorbs the heat load and creates a vacuum environment and an accurate metrology frame to create accurate mechanical interfaces all specifications are met. And a design is made for an accurate and stiff repeatable interface for the Collector Module.

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- [1] HertzWin 1.2.1, Vink System Design & Analysis, 2009.
- [2] Ioan Cracaoanu and Fridtjof Bremer, Philips Innovation Services, APT536-11-0466, 2012

Silicon Carbide Joints for Thermo-Mechanically Stable Assemblies

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OVERVIEW

Multi-component ceramic arrays often depend on the ability to join various ceramic components into reliable high strength systems. Applications include metrology tools and stages, precision optics, focal plane arrays, and electronics wafer handling stages (see Figure 1). These applications are sensitive to thermal gradients that create displacements in assemblies that can limit the accuracy desired for precision processes. In complex systems, the size and number of sub-components require joints where materials and thermal conditions can be discontinuous. These dissimilarities can result in residual stresses and/or thermal discontinuities. This research discusses the strength of various joining materials, including a newly developed joining material that meets the strength of monolithic silicon carbide, and also minimizes thermal discontinuities allowing complex, thermo-mechanically stable assemblies to be constructed.



FIGURE 1. a) Completed SiC satellite dish. b) Metrology tool assembled from SiC components.

The stability of devices is related to their respective thermo-mechanical properties; in particular, thermal expansion (CTE), thermal conductivity, and its stiffness (Young's modulus). For example, if an assembly was induced to a 0.1 watt heat source, the thermal conductivity defines the internal thermal gradients that develop. These gradients coupled with the thermal expansion define the localized displacements. Assuming a geometrically similar 300 mm component, made of silicon carbide (SiC), alumina, or a super-alloy, the overall length of these components would vary when exposed to the 0.1 watt heat load. If a bolted joint were inserted in the middle, thermal discontinuities exacerbate these displacements. Table 1 indicates the total thermal expansion of these components described in this example. The extreme displacements clearly limit the spatial resolution of precise processes.

TABLE 1. Thermal expansions of 300 mm components exposed to a 0.1 watt heat load.

Material	Expansion (micron)	
	Monolithic	Discontinuous Joint
SiC	1.3	2.8
Alumina	12.3	19.3
Super-Alloy	28.1	28.3

JOINTS AND PROCESSING

For joining SiC to SiC, several joining materials were investigated for strength and corrosion resistance. Joint materials tested include SiC tape-derived interlayer (Figure 2)¹, a polymer derived Polycarbosilane (PCS) interlayer (Figure 3)², and a Ceramtec Proprietary Braze (Figure 4). Both the SiC interlayer and the Ceramtec Braze obtained leak-tight joints, with the PCS-derived joint exhibiting a leak tight nature that is configuration dependent.

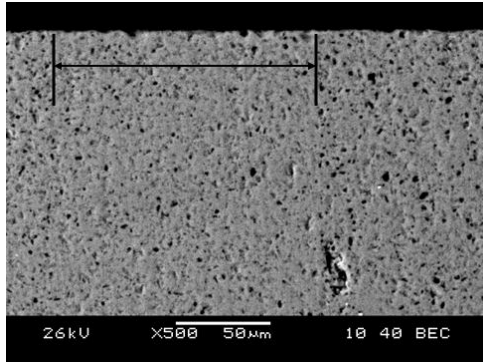


FIGURE 2. SiC-tape derived interlayer. Fired in inert atmosphere to 2100°C.

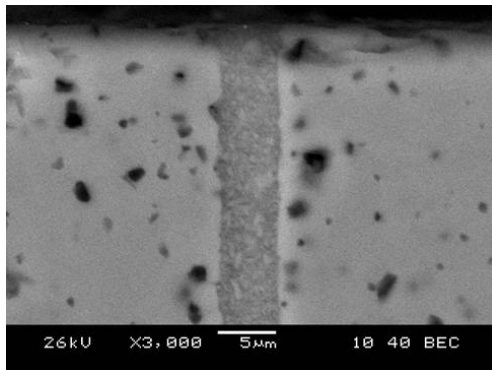


FIGURE 3. Polycarbosilane (PCS) derived joint. Fired in air to 1200°C.

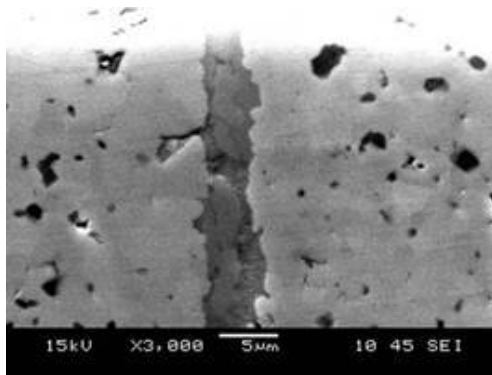


FIGURE 4. Ceramtec Proprietary Braze joint. Fired in air to 1200°C.

Testing of Joints: Double Lap Shear

Three 1" x 1" x 1/4" thick SiC tiles (Figure 5a) were joined using each of the joining materials discussed previously. These triple tile joints were then machined to smaller notched testing specimens (Figure 5b). FEA analysis (Figure 6) shows that this particular notched specimen geometry allows for shear testing of the joints.

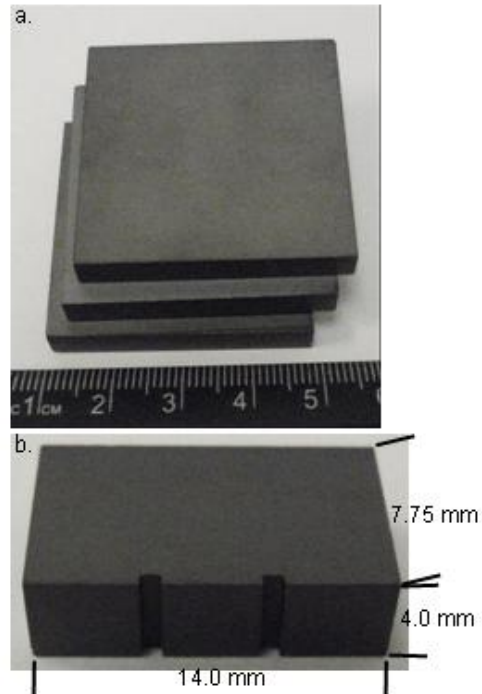


FIGURE 5. a) SiC tiles before joining. b) Joined tiles machined into smaller double lap shear specimens for testing. Joint region is notched.

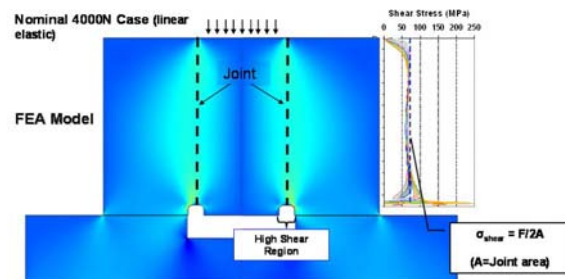


FIGURE 6. FEA model showing shear in the joint region.

Joined and notched specimens were tested for shear strength at both room (22°C) and elevated temperatures (500°C, 750°C, and 1200°C) in an air ambient within a high temperature furnace (Figure 7). Additional samples were also exposed for 1000 hours in high temperature (500°C and 1000°C) environments of N₂ in one case, and a 50/50 vol% mixture of air/H₂O in

another case. The test rig for exposure is shown in Figure 8. These exposed samples were re-tested in shear at room temperature to determine the effects of the exposure on shear strength.

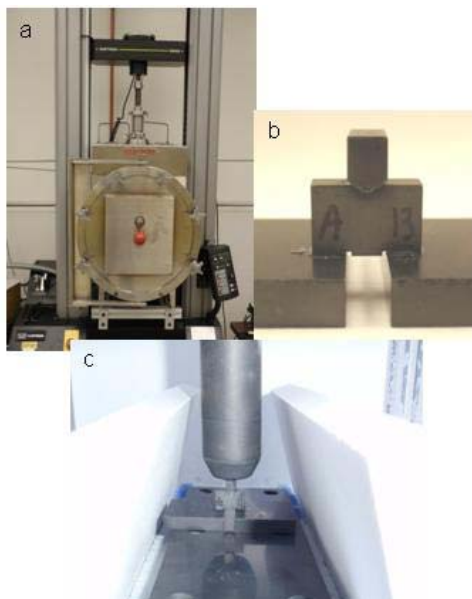


FIGURE 7. High temperature double lap shear testing: a) samples are loaded into furnace, b) specimen set-up for shear testing, c) loading of specimen. Testing is then completed at the desired temperature, with the furnace itself previously placed into an Instron load frame, (Instron testing machine, Model 5566, 10 kN static load cell).



FIGURE 8. Cross-section of the high temperature corrosion furnace used for specimen exposure. Individual specimens are loaded within quartz open ended tubes.

RESULTS AND DISCUSSION

In-use stresses cause shear at the joint due to thermal gradients and differential loads. Testing of materials are often done using 4-pt bend

rather than shear testing. Shear strengths will be lower than strengths obtained in 4-pt bending. Figure 9 shows the resultant room temperature shear strengths of these joined samples. For reference, the 4-pt values for the same joints are also shown. Here is noted that the Ceramtec Braze strength is near monolithic strength values of SiC.

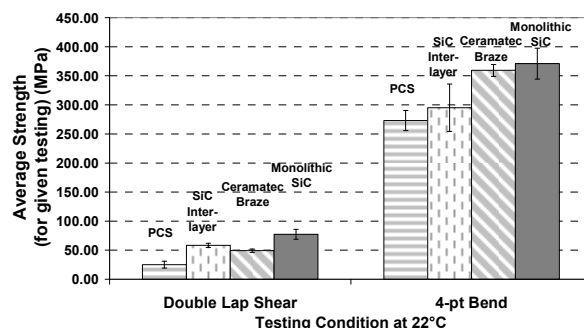


FIGURE 9. Room temperature strength vs testing method for joined specimens.

It was found that in stronger joints, initial failure would begin near the joint (in the notch region), then in an arcing pattern, fail through the bulk of the SiC – see Figure 10. Finite element analyses indicated that this pattern follows a path that is perpendicular to the maximum principle stresses; suggesting that the joint strength is similar to the bulk strength of the SiC. Lower strength joints (such as PCS) show that the majority of failure following the joint itself.

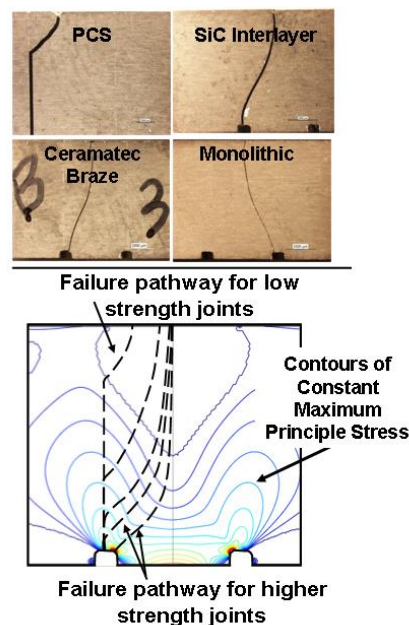


FIGURE 10. Failure patterns for double lap shear samples and finite element results.

The results for the high temperature shear testing are shown in Figure 11. Note that here the room temperature shear test results are those seen in Figure 9 for reference.

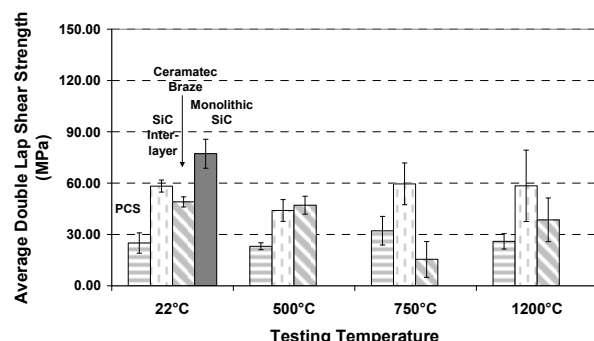


FIGURE 11. Double lap shear strength vs testing temperature for selected joining materials.

From Figure 11 we see that the PCS and SiC interlayer joints are relatively insensitive to temperature, with the Ceramtec Braze also being insensitive at temperatures at and below 500°C. The Braze joint strength decreases above 500°C but recovers strength as temperatures increase due to other phases forming within the braze.

Samples that underwent high temperature exposure, as noted previously, were then tested in shear at room temperature. The results are shown in Figure 12.

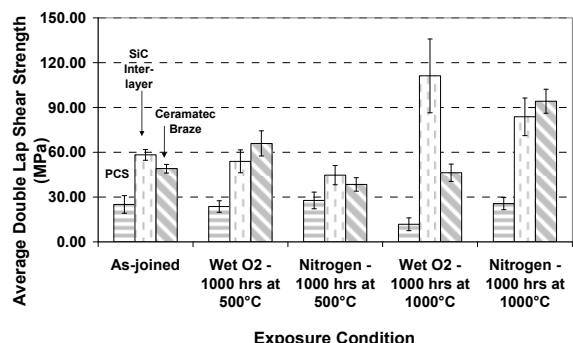


FIGURE 12. Shear strength vs. exposure condition for selected joints.

From Figure 12 we see the following: The SiC interlayer improves with exposure with the presence of oxygen on the surface of the joint (Figure 13a, b), with the increase in oxygen (a glassy phase) being confirmed by EDS. The PCS joint is stable except at 1000°C in wet oxidizing conditions, where microscopy reveals joint recession (Figure 13c,d). The Braze joint shows phase evolution during the exposure

process (Figure 13e,f), with those mechanisms currently being investigated.

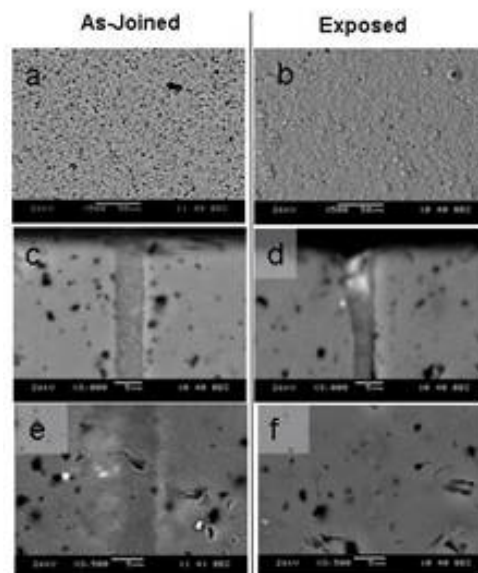


FIGURE 13. SEM micrographs showing joints before and after exposure for 1000 hours at 1000°C.

CONCLUSIONS

For PCS derived joints, the shear strength is about 35% of monolithic SiC values. This joint is stable with temperature and in tested environments, except in wet oxygen at 1000°C. SiC interlayer joints have shear and bend strengths that are near monolithic SiC values (about 80%). These joints are stable with temperature and oxidizing environments; however they are expensive in processing. The Ceramtec Braze joint has bend strengths that are equivalent to monolithic SiC. However, the shear strengths are somewhat reduced compared to the monolithic values, but improve with exposure with the joint strength increasing as the joint material evolves. Further study is ongoing to determine the interactions of this joint microstructure and their effects on joint strength.

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Dual Moiré and Laser Interferometry as a Metrology Tool

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INTRODUCTION

Measurement of physical parameters of a large object or scene accompanies by losing detailed information about small regions of object. In fact techniques developed to make measurement in large scale (macro) are very different from those measuring at small scale (micro). This problem is one of the challenges in the field of surface metrology.

One optical method that is used for surface metrology at macro scale is Moiré technique [1]-[5]. Laser interferometry is another metrology method which has been used for measuring at micro level [6][7]. With this method one can get the resolution of up to sub-nanometer for nearly spherical or flat surface [8], but is very challenging if not impossible to apply to arbitrary geometries. There is no reported work that combines the large scale measurement of Moiré techniques with high accuracy of laser interferometer.

This article addresses exactly this challenge. The work presented in this article combines these two techniques allowing detailed surface texture of a small region of a very large object to be analyzed while the overall form and shape of the object is measured.

We designed and implemented a dual Moiré and laser interferometer system. The system was designed and simulated with MATLAB software. The dual Moiré and laser interferometer then was implemented and the interferogram from both Moiré part and laser interferometer part were acquired.

The challenge here is to stitch data from Moiré system and laser interferometer. Authors proposed a method [9] based on image

registration to accomplish stitching. An algorithm based on proposed method was developed to combine the data from Moiré and laser interferometer and MATLAB software was developed to analyze the fringe pattern from Moiré and laser interferometer. At the end, an uncertainty analysis was performed to characterize the implemented system.

The rest of the article organized as follows: First, system design and implementation is described. Then the simulation is explained. The stitching algorithm is given afterward. In the next section, the experiment will be demonstrated and in the last section the article will be concluded.

SYSTEM DESIGN

Figure 1 shows the schematic of proposed system. For Moiré measurement, fringe projection technique is used. The Moiré pattern is projected from the left arm with an angle. To

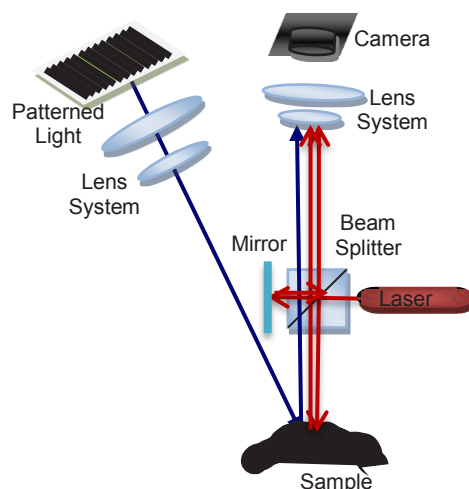


FIGURE 1. Schematic of Dual Moiré and laser interferometer

reduce the perspective effect, a telecentric lens is used as a part of lens system in front of projection part.

The laser interferometer is a Michelson interferometer. The Michelson interferometer is placed close to object and in front of optical elements. This choice of location of laser interferometer makes sure that the laser wavefront almost only deformed by the object and not optical elements before interfering with reference beam of the laser. This is important, because compensating for wavefront change that causes by optical elements is very difficult, in software, if not impossible task.

Figure 2 shows the implemented system. A projector is used in fringe projection system to project Moiré patterns. Using projector gives the flexibility of changing the Moiré patterns in software other than hardware. Therefore the equivalent wavelength of projection Moiré and also the color of pattern can be changed in software.

SIMULATION

To evaluate the system, both fringe projection and laser interferometer of the system were simulated in MATLABTM environment. For fringe projection, the effect of finite resolution of camera was considered and the interferogram was generated. Moiré simulation was performed using ray-tracing technique.

For laser interferometer, the effect of noise and speckle were considered in simulation and interferograms were generated for some typical surfaces. Laser interferometer was modeled as

$$I(r, t) \approx I_0 [1 + V \cos \delta(r)] R_s(r) + R_E(r, t)$$

where V is fringe contrast, $\delta(r)$ is phase that corresponds to profile of the object, $R_s(r)$ is speckle noise and $R_E(r, t)$ is electronics noise.

Parameters of simulations are as follows: speckle noise is considered as averaged exponential distribution; electronics noise is additive white Gaussian noise (AWGN) with signal to noise ratio (SNR) of 20dB and contrast was one. The angle between camera arm and projector arm is 20 degrees; camera is 512 by 512 pixels; laser wavelength is 633 nm; number of fringes at lowest zoom level for projection Moiré is 51.2 and the grating pitch of Moiré pattern is .1 mm.

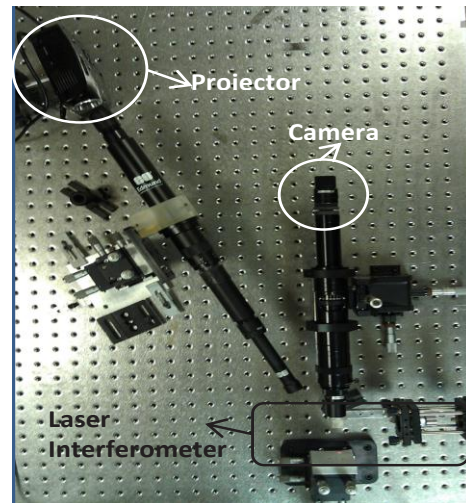


FIGURE 2. Dual Moiré and laser interferometer

The simulated object has both form and features. The form of the object is a circular symmetric cosine shape with amplitude of 0.2 mm and period of 2 mm. Features are also circular symmetric cosine shape with amplitude of 1 μm and period of 2 μm .

Figure 3 shows the simulated object at different zoom levels and the corresponding simulated interferograms for both fringe projection and laser interferometer. As it is seen, at lower zoom level the Moiré pattern are visible and the form of the object can be constructed from the deformed pattern. The laser interferogram, on the other hand, does not have a pattern such that phase map can be extracted. At higher zoom level, the laser interferogram has meaningful information, i.e. the phase information can be extracted from laser interferogram and consequently features of the object will be reconstructed, while the Moiré pattern does not have significant information.

STITCHING ALGORITHM

After acquiring interferograms from fringe projection and laser interferometer, interferograms should be stitched together to maintain a continuous image of the object. Authors have been developed a technique to calculate the location of all acquired interferograms relative to a reference point on object [9]. To do so, first an image is acquired at lowest zoom level using fringe projection. Then more interferograms are acquired using both fringe projection and laser interferometer by

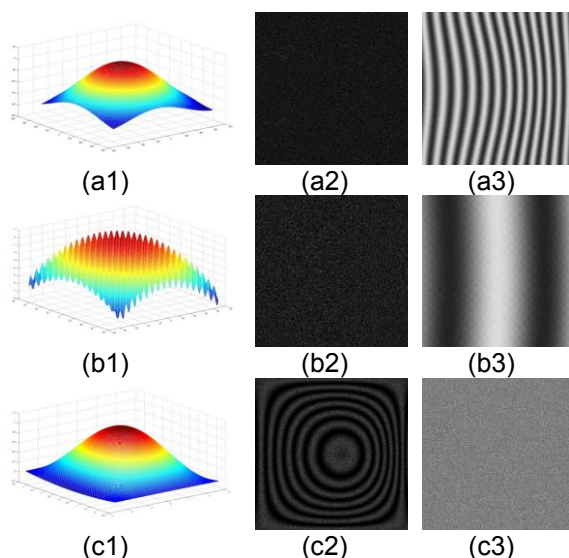


FIGURE 3. Simulated object and the resultant interferogram from fringe projection and laser interferometer at different levels of zoom: (a1): object at zoom 2x, (a2): laser interferogram at zoom 2x, (a3): Moiré pattern at zoom 2x, (b1): object at zoom 64x, (b2): laser interferogram at zoom 64x, (b3): Moiré pattern at zoom 64x, (c1): object at zoom 512x, (c2): laser interferogram at zoom 512x, (c3): Moiré pattern at zoom 512x.

zooming in on regions of interest. Using our technique, the exact zoom level change (scale) and shift in lateral directions (translation in x and y axis) will be calculated. Therefore the coordinate of each point at any interferogram is available relative to a reference point on first interferogram at lowest zoom level.

Figure 4 shows the results of applying that technique to simulated interferograms. Since only one camera is used for acquiring interferograms from both fringe projection and laser interferometer, the pixels of acquired interferograms from fringe projection and laser interferometer are corresponds to each other. Therefore in a specific zoom level, switching from fringe projection to laser interferometer (or vice versa) does not require further calculations to correspond pixel coordinates of their interferograms. The result shows sub-pixel resolution for data stitching at different levels of zoom can be achieved.

EXPERIMENT

Figure 2 shows the experimental setup. As mentioned before, fringe projection is used for coarse measurement (form of the object). This is

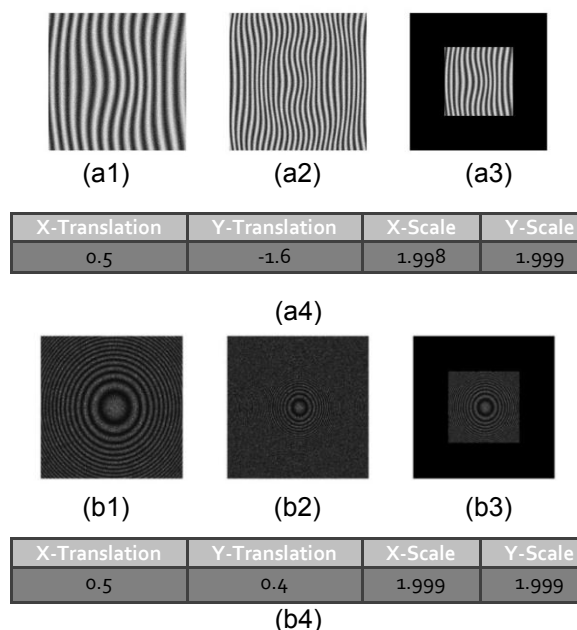


FIGURE 4. Results of image registration to determine the shift and scale in both x and y directions at different levels of zoom. Images (a1) and (a2) are simulated fringe patterns from the fringe projection moiré system at zoom levels of 2 and 1. Image (a3) is the scaled and translated version of (a1). It is located with respect to the coordinate system of image (a2). Table (a4) shows the results of image registration for fringe patterns from fringe projection. Images (b1) and (b2) are simulated fringe patterns from the laser interferometer at zoom levels of 256 and 128. Image (b3) is the scaled and translated version of (b1). It is located with respect to the coordinate system of image (b2). Table (b4) shows the results of image registration for interferogram from laser interferometer.

done to generally inspect the whole area of the object and to find the regions of interest.

Laser interferometer part of the system is used for fine measurement. Interferograms from laser interferometer are captured and combined with Moiré fringes at different levels of zoom. It is done by the means of dynamic range of zoom lens and objective lens put back-to-back.

Figure 5 shows the acquired fringe patterns and the final stitched image. Stitching fringe patterns was performed by using the image registration technique [9].

To characterize the system, an uncertainty analysis was performed for both Moiré and laser interferometer, namely the repeatability of the Moiré and laser interferometer were measured.

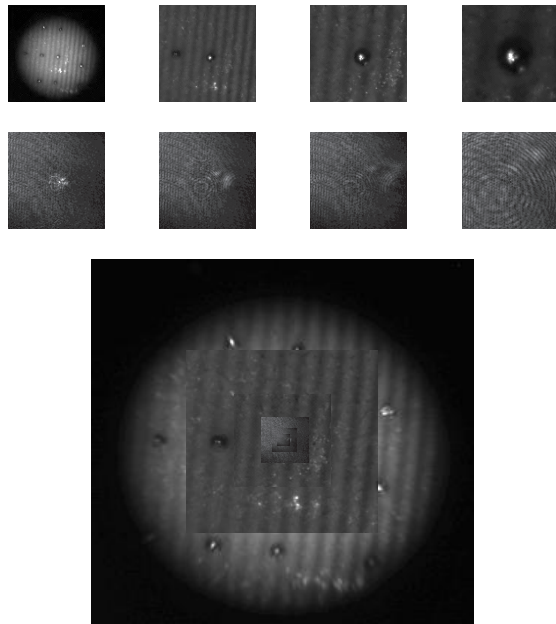


FIGURE 5. Interferograms from fringe projection (first row) and laser interferometer (second row) at different levels of zoom and the stitched image (third row).

Results of uncertainty analysis are given in Table 1. As it is shown, the error for fringe projection is below 4% for fringe projection which is typical for a Moiré system. The error for laser interferometer is below 1% for different zoom level.

The majority of error for measurements comes from instability of laser wavelength, vibration of optical table, air flow and electronics noise. Also there are diffractions that are represented as dense circular fringe patterns that contribute to error.

CONCLUSION

We designed and implemented a dual Moiré and laser interferometer system. The system was designed and simulated with MATLAB software. The dual Moiré and laser interferometer then was implemented and the interferogram from both Moiré part and laser interferometer part were acquired.

Results show that the dual Moiré and laser interferometer is capable of measuring the object in both low and high resolutions and the data from both Moiré and laser interferometer can be stitched to construct a continuous image of the object under measurement.

TABLE 1. Error percentage at different zoom levels for fringe projection and laser interferometer

Fringe Projection		Laser Interferometer	
Zoom Level	Normalized Error %	Zoom Level	Normalized Error %
1	0.36	3.75	0.106
1.5	0.44	5	0.372
2	0.5	7.5	0.541
3	1.39	10	0.977
4	3.81	15	0.518
6	3.34	20	0.627
8	4.45	30	0.592
10	5.57	32.5	0.787

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DEFORMATION ERROR OF EPOXY-REPLICATED X-RAY MIRROR

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INTRODUCTION

Grazing incidence X-ray optics is based on the total reflection of X-rays from a smooth surface. One well-known grazing incidence optic is a Wolter type-I mirror [1]. Despite its geometry being very well studied, this aspheric mirror was not widely used until the middle 1970s because of its tight surface and figure tolerances [2]. An X-ray mirror of 40 mm in diameter for a laser fusion study was manufactured by direct single-point diamond turning (SPDT) and polishing of the reflecting surfaces. However, an X-ray mirror for microscopic study is difficult to manufacture by direct machining of the inner reflecting surface, because the mirror has a small diameter (less than 20 mm) and a large length (larger than 30 mm), even though ultraprecision machines and fabrication technologies have progressed much. In this case, a replication method [3–5] for fabrication of the X-ray mirror is much better.

A replicated mirror can be fabricated from a master mandrel, which has a counter shape of the desired mirror, through several replication steps. A glass replication method [3] has been well established for the fabrication of soft X-ray mirrors. However, the replication method has several disadvantages. One critical problem is that the replicated surface is imperfect [6]. In other words, there is a large difference between the master mandrel and the replica in figure and surface roughness. The accurate shape and ultrasurface are highly recommended for improving focusing performance and getting high X-ray reflectivity in X-ray microscope mirror. Moreover, one master mandrel can be used to fabricate only one replicated mirror.

In this paper, we examine the deformation error in terms of surface roughness and figure of a flat X-ray mirror fabricated from epoxy replication method in stead of glass replication technique.

EPOXY REPLICATION

The fabrication process of the epoxy replication has four steps. The first step is to prepare the master mandrel, molding die. The smooth surface and accurate shape of the molding die is very important because the replicated mirror critically depends on the surface quality of the molding die.

The next step is coating process. Gold was chosen as a coating material. A gold layer of 300 nm in thickness was deposited on the molding die using a DC magnetron sputtering system. The thin gold layer has an important role as the parting agent to separate the molding die and a mirror substrate, and it also serves as a reflecting surface. After Au coating epoxy resin was coated on the molding die. A mirror substrate was put onto the molding die coated with epoxy resin. After curing the epoxy resin, the molding die was released by cooling down, i.e., through the difference of the thermal expansion coefficient between the molding die and the mirror substrate. In this process, the epoxy region may suffer deformation. Through the four steps, the replicated mirror was successfully fabricated without any damage to the reflecting gold surface. Figure 1 shows the fabrication process of the epoxy replication method.

REPLICATED X-RAY MIRROR

A flat molding die of 48.3 mm in diameter was prepared by SPDT of electroless nickel plated on aluminum. The surface roughness of molding die is less than 1 nm rms (root-mean-square) overall area. A replicated X-ray mirror with glass substrate of 3 mm thick was fabricated through the epoxy replication process using the flat molding die. Figure 2 shows the flat molding die and its replica. Several replicas without surface degradation can be made by using only molding die through the epoxy replication process.

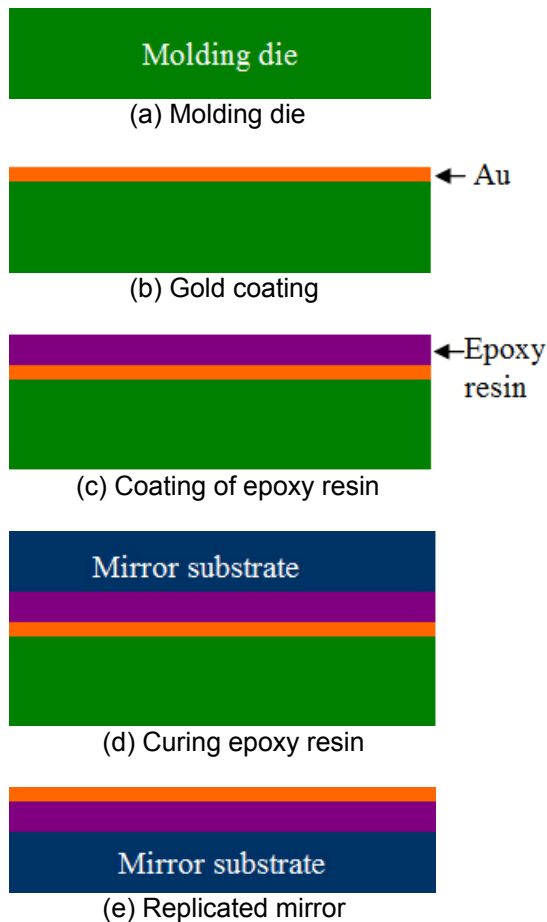


FIGURE 1. Epoxy replication process.

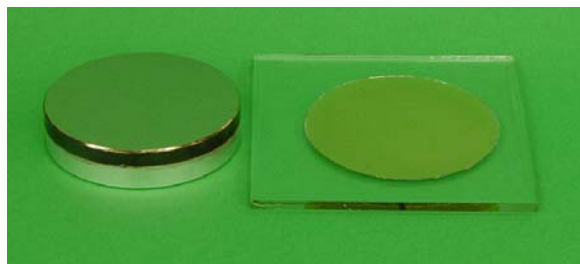


FIGURE 2. Flat master and its replica fabricated by the epoxy replication process.

The shape of the replicated mirror was measured with an optical profiler (ZYGO GPI-XP, USA) in the inverse mode as shown in Fig.3. Peak-to-Valley (PV) and Ra for a replica were 545.2 nm and 103.82 nm, respectively. Those values were very close to those of the master die, PV of 540.25 nm and Ra of 103.60 nm. It meant that two surfaces were very close to each other without making large difference.

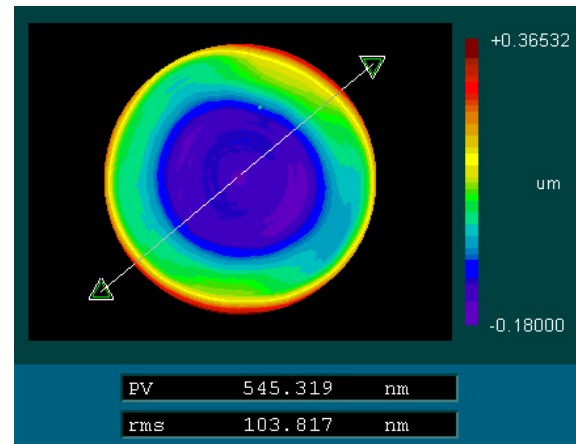


FIGURE 3. Shape of a replica fabricated by epoxy replicated method.

A line to measure a line profile was chosen on the measured topography, and then its counterpart of the master die was carefully searched. Line profiles for the original die and the replicated mirror were compared. Deformation of the replicated mirror was easily found as shown in Fig. 4. The deformation shows almost symmetric shape around the center, vertex. It might come from the symmetric form of the master die. On vertex the deformation was large and maximum deformation was 24.64 nm in the diameter of 48.3 mm. For other line profiles also showed very similar results. In a few tens of millimeter scale, the deformation error of the epoxy resin after all replication process may be a few tens of nanometer scale. This shape difference might contain the deformation and shrinking of the epoxy resin.

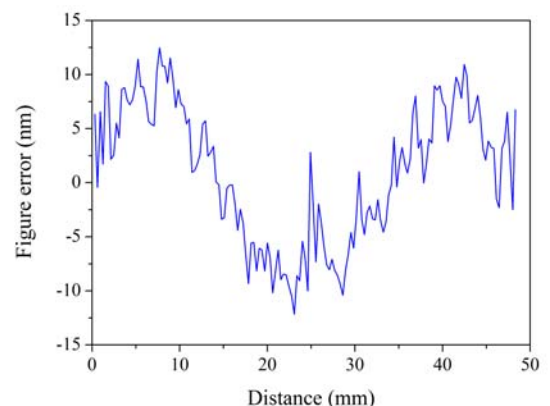
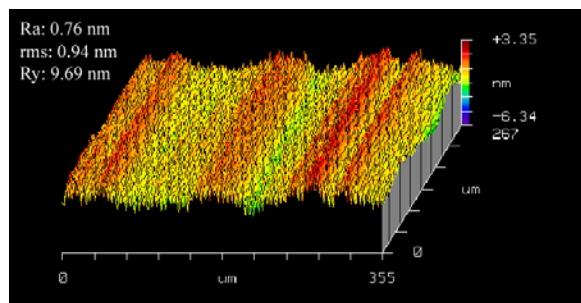


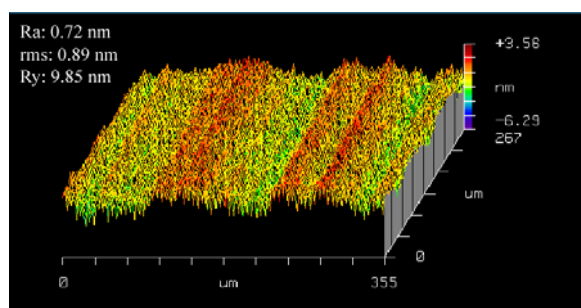
FIGURE 4. Shape deformation of the replica mirror.

The surface roughness of the replicated mirror was measured with an optical profiler (ZYGO

New View 200 HR, USA) with 20X magnification objective. The measurement area was $360 \times 270 \mu\text{m}^2$. Two surface topographies for the replica and its master die were very similar and surface roughnesses for each other were also very close as shown in Fig. 5.



(a) Replicated mirror

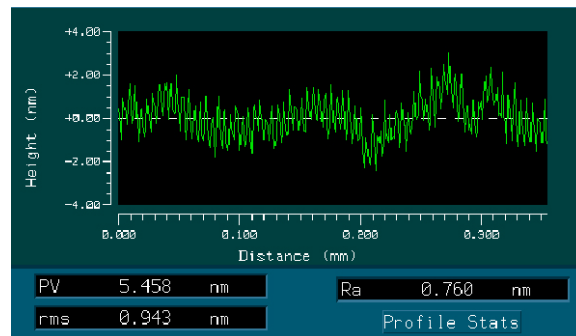


(b) Molding die

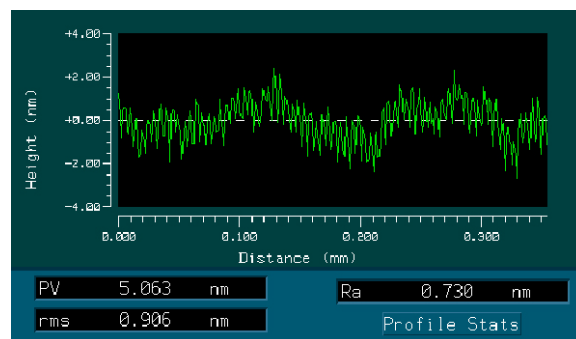
FIGURE 5. Surface topographies of the replicated mirror (a) and its molding die (b).

For the measured area, line profiles were compared to examine the deformation of epoxy after replication. PV, Ra and rms values were very similar as shown in Fig. 6. Two surface topographies of the molding die and its replica were not completely the same. In other word, the replicated mirror had very small deformation of less than 5 nm. This deformation could occasionally improve the surface roughness compared to the original molding die.

Most area of the replica mirror showed some defects as shown in Fig. 7, and many tiny peaks on the master die were also observed after the separation step. The peak defects on the replica mirror were made during separation step in the replication process. The height of peaks was ranged approximately from 2 nm to 6 nm. These peaks might be related to the Au coating on the master die. Many small grains of Au particles with various heights were found on the thin gold layer.



(a) Replicated mirror



(b) Molding die

FIGURE 6. Line profiles for the replicated mirror (a) and the molding die (b).



FIGURE 7. Surface image take with a Nomarski Microscope.

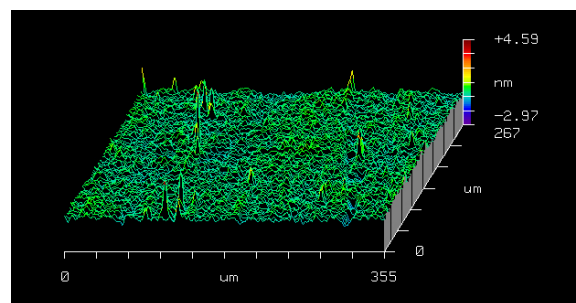


FIGURE 8. Surface topography with peaks on the replica mirror.

CONCLUSIONS

The surface deformation was examined for the X-ray mirror fabricated by the epoxy replication method. The deformation errors of epoxy-based mirror were less than 5 nm and 25 nm within a line of 355 μm (a few hundreds of micrometer) and a diameter of 48.3 mm (a few tens of millimeter), respectively. In addition, defects were founded on the surface of the replica mirror. The height of the defects was ranged from 2 nm to 6 nm.

The epoxy deformation for the replica mirror fabricated by epoxy replication method can be disregarded for soft X-ray focusing mirror. The surface errors for the X-ray mirror fabricated by the epoxy replication process can give information when X-ray mirror is designed for specific applications.

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EXPERIMENTAL EVALUATION OF PHASE DIGITIZING NONLINEARITY CORRECTION IN HETERODYNE INTERFEROMETRY

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INTRODUCTION

Frequency leakage in polarization-dependent heterodyne interferometers results in nonlinearity, or periodic error, in the phase digitized result. Phase digitizing is the process of converting the interferometer signal, obtained from the interference of the reference and measurement beams, into a digital representation of phase. In a previous approach [1-5], the periodic error is removed in a two-step process including phase digitizing, and second, first order periodic error (i.e., a cyclic error with one cycle per wavelength of optical path change) removal from phase digitized data. The second step is repeated for second order periodic error (with two cycles per wavelength of optical path change). This paper describes experimental results for a method where the first order periodic error is removed in the phase digitizing process in a single step. The accuracy of this measurement is not sensitive to target velocity, provided a minimum velocity threshold is at least temporarily exceeded. Because the latency requirement is much reduced from the two-step process, the achievable accuracy in measurement and compensation is improved.

PERIODIC ERRORS IN HETERODYNE INTERFEROMETRY

Figure 1 shows a schematic representation of a heterodyne setup for a single-pass configuration. In the ideal case, the two collinear, orthogonally-polarized laser frequencies (f_1 and f_2) are perfectly split at the polarizing beam splitter and are directed into the measurement and reference arms of the interferometer. The measurement arm contains the moving retroreflector mounted on the stage whose displacement is to be measured, while the reference arm retroreflector is stationary. Motion of the moving retroreflector causes the measurement arm frequency to be Doppler shifted by f_d . The two frequencies from the

measurement and reference arms then pass through a mixing linear polarizer (to cause interference) and are collected by the photodetector, which carries the interference signal to the phase measuring electronics. This Doppler-shifted measurement arm frequency is compared to a reference interference signal in the phase measuring electronics and is used to determine the displacement information. However, due to non-ideal performance, leakage of each frequency into both the measurement and reference arms can occur. This frequency leakage gives rise to nonlinearity in the displacement signal. This nonlinearity is often referred to as periodic error, which is non-cumulative in nature and repeats with integer wavelength changes in the optical path.

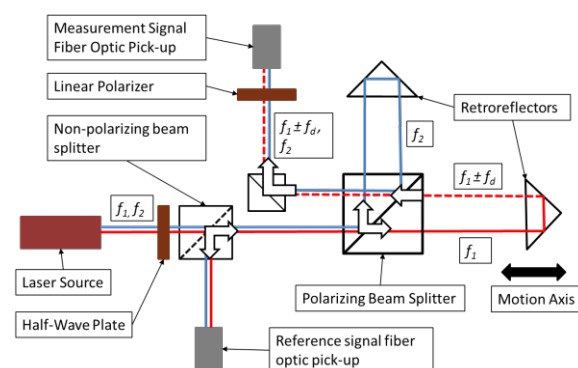


FIGURE 1. Interferometer setup with frequency leakage.

EXPERIMENTS AND RESULTS

Nonlinearity Correction (Constant Velocity)

Experiments were conducted to verify the nonlinearity correction. Initially, tests were conducted at constant velocity, v , stage motion. The randomly selected misalignments of the half wave plate and linear polarizer resulted in a mean first order periodic error of 8.2 nm and mean second order periodic error of 2.2 nm. Figures 2 and 3 show the first and second order periodic errors for different velocities. A

comparison of the results with and without the nonlinearity correction enabled is provided. Figure 2 shows that at stage velocities which produce a Doppler shift magnitude greater than 40 kHz (i.e., the stage speed is greater than 759.6 mm/min) there is a significant reduction in the first order periodic error magnitude. In the current configuration, the nonlinearity correction was not enabled when the Doppler frequency was less than 40 kHz. Because the periodic error correction algorithm implemented in this study does not address the second order component, there is little effect of enabling the nonlinearity correction; see Figure 3.

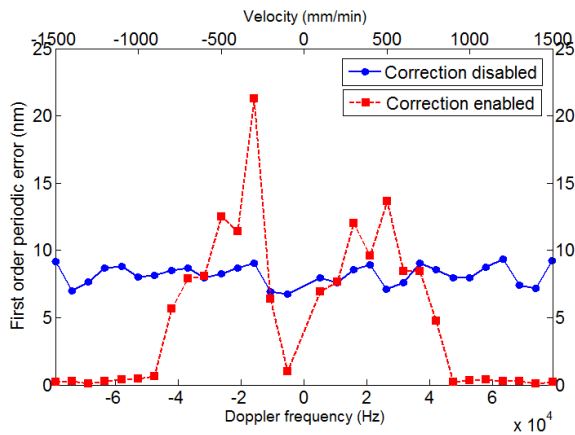


FIGURE 2. First order periodic error.

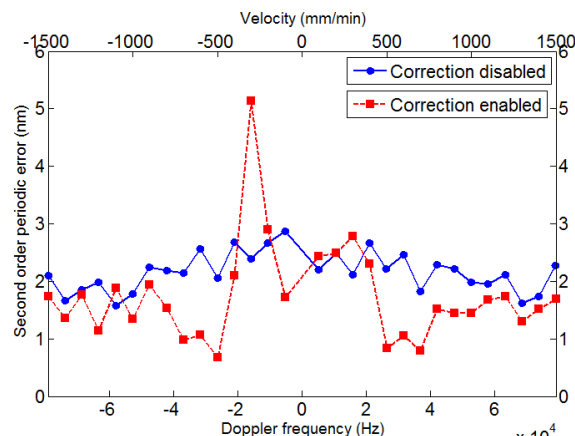


FIGURE 3. Second order periodic error.

Nonlinearity Correction (Sinusoidal Motion)

Tests were also conducted for non-constant velocity. The stage was commanded to move in a sinusoidal motion with a low range and high frequency. This ensured that the complete sinusoidal cycle was captured within a relatively small sampling time (32000 samples at a sampling rate of 312.5 kHz). The collected data was sectioned into small time intervals (0.00125

s, arbitrarily selected). Each section of the data was processed individually. The position data was converted to a stationary signal in the position domain using interpolation and the first and second order errors were then calculated for each section [6]. This technique is discussed in detail later. The results reported here are for the case when the stage was commanded to move in a sinusoidal motion with a total range of 0.4 mm at a frequency of 20 Hz.

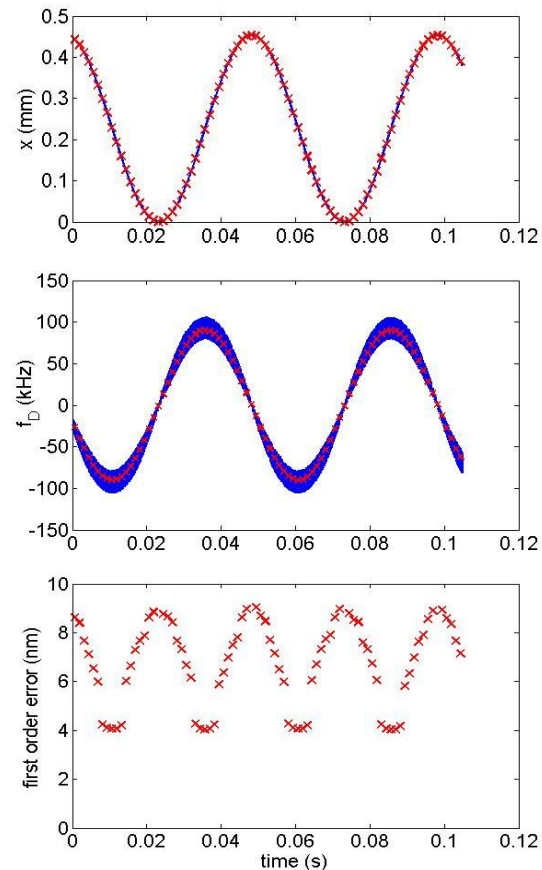


FIGURE 4. Position (x), velocity, and first order periodic error. Nonlinearity correction disabled.

Figure 4 shows the position, velocity, and first order periodic error magnitude as a function of time with the nonlinearity correction disabled. The second order periodic error was not analyzed. The measured motion amplitude is higher than the commanded amplitude. This could be due to overshoot and acceleration limitations of the stage. The red crosses indicate the calculated mean value of position and velocity for each section. The velocity was converted into a Doppler shift frequency using:

$$f_d \text{ (Hz)} = v \text{ (mm/s)} * (2/633) * 1 \times 10^6.$$

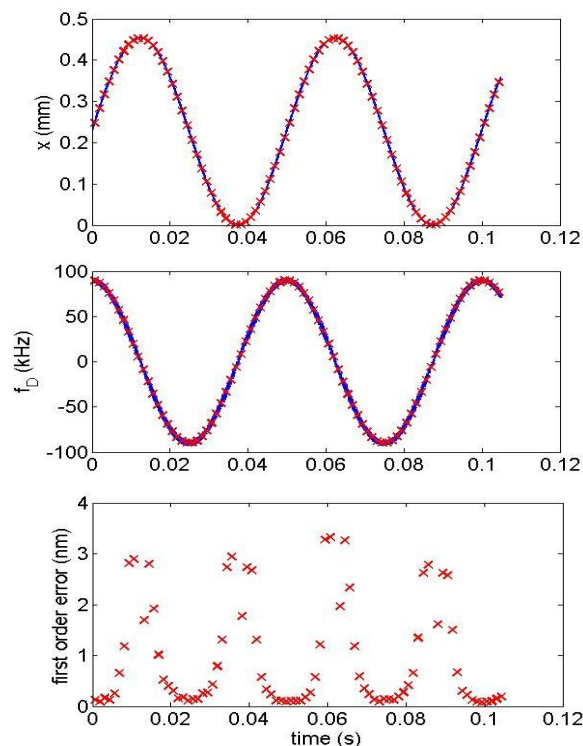


FIGURE 5. Position, velocity, and first order periodic error. Nonlinearity correction enabled.

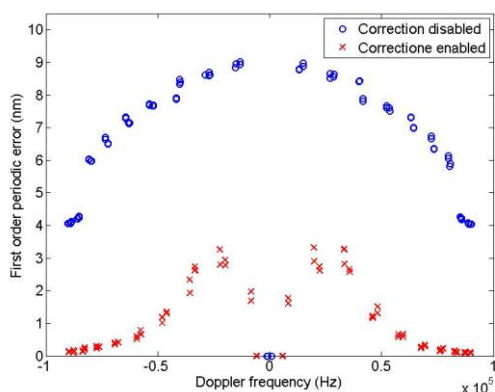


FIGURE 6. First order periodic error as a function of velocity (Doppler frequency shift).

Figure 5 shows the results for the same experimental conditions with the nonlinearity correction enabled. A decrease (improvement) in the first order periodic error is observed. However, at low velocities the error corrections are worse than at higher velocities. These are at the end of the oscillatory motion where the stage is reversing direction. In Figure 6 the first order periodic error is displayed as a function of the velocity (Doppler frequency shift). A considerable improvement in the error was

observed when the nonlinearity correction was enabled.

Position Domain Evaluation of Periodic Error

Each point (marked by the red 'x') in Figures 4 and 5 was obtained by partitioning the data into equal sections in the time domain and analyzing each section individually. The process of determining periodic error is discussed here. Figure 7a shows the total velocity profile and the cross indicates the section selected to illustrate the technique used to identify first and second order periodic errors. Figure 7b shows the motion profile of the selected section. A polynomial fit was applied to the position data. This polynomial fit was then removed from the raw position data to isolate periodic errors.

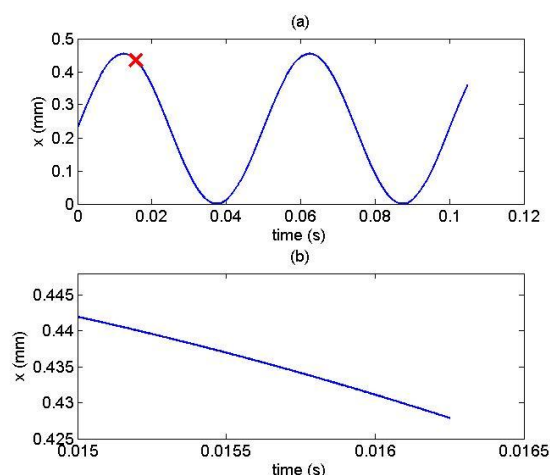


FIGURE 7. Total motion profile. 'X' identifies the selected section (a); the motion profile for the selected section (b).

Figure 8a displays the periodic errors for the selected section. Note that the horizontal axis now denotes position and the error is plotted as a function of position change. This error data was interpolated at equal positional increments, thereby providing equally spaced data in the position domain. Figure 8b shows the interpolated data. Note that only a small fraction of the data in 8a has been plotted in 8b (change in horizontal axis scale). The Fourier transform of this error data in the position domain was then calculated. Figure 8c shows the result, where the horizontal axis was normalized to identify the harmonics of the change in the optical path difference in the interferometer. The first and second order errors are easily identified.

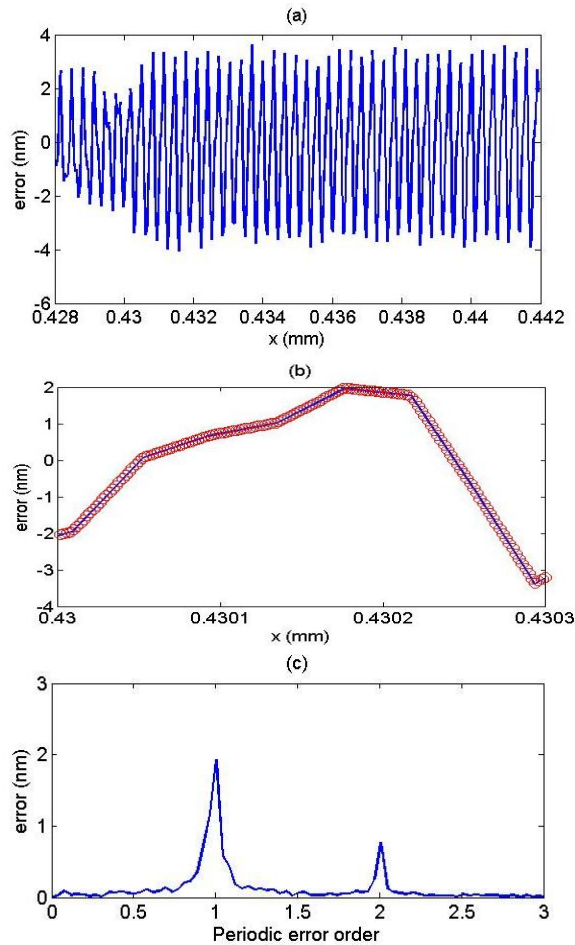


FIGURE 8. Periodic error as a function of position (a); interpolated periodic error (one cycle only) (b); Fourier transform of periodic error (c).

DISCUSSION

In these experiments, the nonlinearity measurement was automatically inactivated when the measured instantaneous Doppler shift magnitude was less than 40 kHz. This was necessary because at a zero Doppler frequency both the intended interference signal and the periodic error components occur at the same frequency, which makes their separation impractical using this algorithm. The exact choice of 40 kHz deserves further investigation because it represents a tradeoff. A larger value allows for a better worst case scenario (when the stage slows to zero velocity after previously travelling at non-zero velocity), but it also reduces the overall average accuracy of the correction. In a system that travels at non-zero velocity, but then crosses below the 40 kHz boundary, nonlinearity measurement would cease, but correction would still continue given

the previously measured parameters. Because the 800 mm/min data is near the limiting velocity (759.6 mm/min), interesting results are observed (Figure 2). Even though the true velocity varied only slightly due to the large periodic error magnitudes, the measured velocity (Doppler frequency) often fell below the cutoff frequency for nonlinearity measurement.

CONCLUSIONS

This paper presented experimental results for a new phase digitizing nonlinearity correction algorithm. It was demonstrated that periodic error was significantly reduced, provided the limiting Doppler shift magnitude of 40 kHz was exceeded. Similarly, for sinusoidal motion profiles it was shown that the correction in periodic error was better at higher velocities and was the worst at the positions where the stage reversed its direction (Figure 5).

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STUDY ON A 1D HOMODYNE ENCODER AND A NOVEL 2D HOMODYNE ENCODER

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INTRODUCTION

Precision displacement measurement is one of the key enabling technologies for precision manufacture which is playing a more and more important role in our modern society. At present, laser interferometers and interferometric encoders are the main solutions to the displacement measurements which require up to hundreds of millimeters of measurement range and nanometer order of measurement uncertainty. Both solutions have their own advantages. But due to the fact that the requirement on measurement performance is increasing, in particular the requirement on the measurement uncertainty is becoming more and more challenging. This is especially true for the metrology sensors of integrated circuits manufacturing equipments. Therefore, in non-vacuum environment, encoders are becoming more and more popular due to their lower sensitivity to fluctuations of refraction index of ambient air and the wavelength variations of their light sources, compared to interferometers [1].

In addition, for applications like some linear motors which need out-of-plane measurement to compensate motion error [2], 2D (one is in-plane another one is out-of-plane) measurements are demanded.

To these ends, in this work, first a 1D homodyne encoder was built and its periodic nonlinearities and noise were investigated, since periodic nonlinearity is an important systematic source of measurement uncertainty. Then a novel 2D homodyne encoder was put forward and its preliminary proof-of-principle experiment was performed.

1D HOMODYNE ENCODER

Design

Figure 1 shows the schematic of the 1D homodyne encoder. This 1D encoder has three measurement channels, but all the three channels measure the same displacement (in X direction) with almost the same performance, which was proved experimentally. But in this paper only the channel 1 will be discussed. In this encoder, head grating 1 and 2 have the same grating period which is half that of grating scale. Both head grating 1 and head grating 2 are placed in the Littrow configuration with respect to the grating scale.

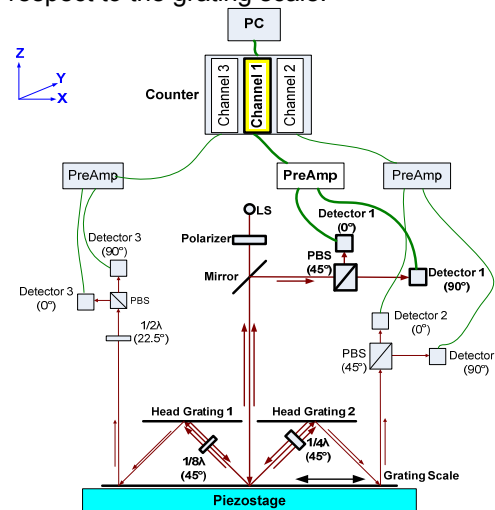


FIGURE 1. Schematic of the 1D homodyne encoder.

As shown in Figure 1, the collimated laser beam is directed perpendicularly to the grating scale with an out-of-plane angle 1.6° , after passing through a polarizer. Then following diffraction on the grating scale, part of the $+1^{\text{st}}$ diffraction order beam is diffracted back to the grating scale again by head grating 1 along the same path and passes a $1/8\lambda$ wave plate twice. Similarly, part of the -1^{st} diffraction order beam is diffracted back by the head grating 2 with double

passages through a quarter wave plate. Then both beams are diffracted by the grating scale again, and the secondary -1st diffraction order of the former +1st diffraction order beam and the secondary +1st diffraction order of the former -1st diffraction order beam are overlapped and interfere. The interfering beams carry the phase information, which is introduced by the displacement of the grating scale. Finally the interference signals are detected by detector1. And the phase information is processed and interpreted by the Data Acquisition and Processing Electronics (DAPE).

Comparison Measurement with A Differential Interferometer

To investigate the performance of this 1D homodyne encoder, a comparison measurement with a differential heterodyne interferometer [3] was performed. The comparison measurement setup is shown in Figure 2. The grating scale together with the interferometer measurement mirror is fixed on a custom PI six-axis piezo stage and the interferometer reference mirror is fixed at the encoder head. During the measurement the piezo stage is continuously moving in X direction back and forth in a triangle waveform. This movement is measured by the 1D homodyne encoder and the interferometer which are synchronized by a continuous external Trigger. The period of grating scale is 1200 grooves/mm and the wavelength of laser is 532 nm. The nominal periods of the 1D encoder and the interferometer are 208.333 nm and 266 nm respectively.

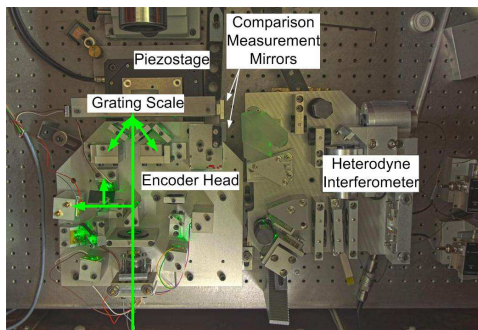


FIGURE 2. setup of comparison measurement with a differential interferometer.

During this comparison measurement, two kinds of DAPE's were used; one was the Heidenhain EIB 741 another one was a phase meter based on fast ADCs [4]. The latter one employs 16-bit flash ADCs with signal-to-noise ratio of 80 dB and sampling rate of up to 100 MHz, which

result in a higher resolution and higher sampling rate compared to EIB 741.

When EIB 741 was employed, the built compensation-for-position-value function was enabled. While the phase meter was used, an off-line Heydemann correction [5] was performed. For one file of raw data acquired during the whole comparison measurement time, the correction parameters were calculated only once and applied for all data points. Here periodic nonlinearity results are determined through a Fast Fourier Transform (FFT) of the measurement difference between the encoder and the interferometer.

Results

When the EIB 741 was employed as DAPE of the 1D encoder, the following measurement settings were used:

- Encoder/interferometer synchronized at a trigger rate of 10 KHz;
- Piezo stage moving in triangle waveform with an amplitude of 10 μm at 0.08 Hz;
- Measurement time: 100 s.

The measurement difference between the interferometer and the 1D encoder is shown in Figure 3. The standard deviation of the measurement difference is 117 pm which mainly results from the synchronization error, signal noise, periodic nonlinearities of the two measurement systems and yaw error of the piezo stage. The synchronization error was identified through the relation between the change of the measurement difference and the change of piezo stage moving speed. The synchronization error also can be identified from the amplitude spectrum of the measurement difference.

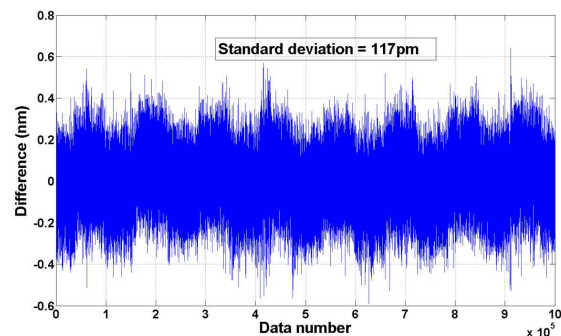


FIGURE 3. Measurement difference between the interferometer and the 1D encoder with EIB 741

Figure 4 shows the amplitude spectrum of the measurement difference, from which a ± 50 pm overall periodic nonlinearity of this 1D encoder is identified. During this experiment the periodic nonlinearity of the interferometer was determined to be ± 20 pm, which results from multiple reflections caused by misalignment.

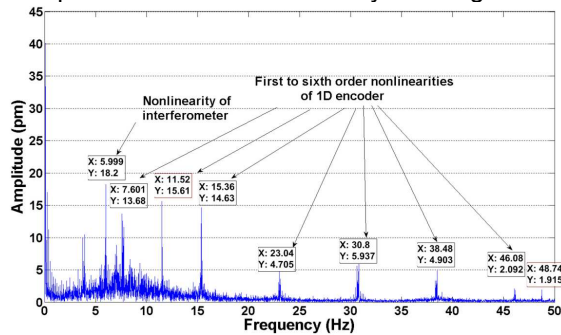
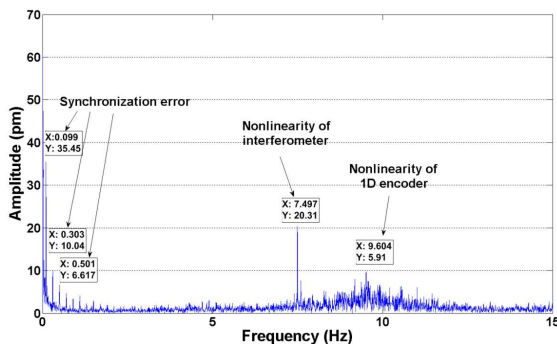
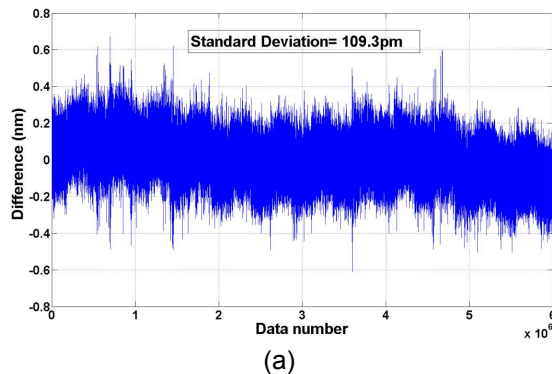


FIGURE 4. Amplitude spectrum of measurement difference (with EIB 741).

When the phase meter was used as DAPE of this 1D encoder, the following measurement settings were used:

- Encoder and interferometer use the same phase meter with data acquisition rate of 48,828 kHz;
- Piezo stage was moving in a triangle waveform with amplitudes of 10 μm at 0.1 Hz;
- Measurement time: 122.4 s.



(b)

FIGURE 5. (a) Measurement difference between the interferometer and the 1D encoder with the phase meter; (b) Amplitude spectrum of (a).

The standard deviations of the measurement difference between the interferometer and the 1D encoder is 109.3 pm, as shown in Figure 5 (a). In Figure 5 (b), the amplitude spectrum of the measurement difference shows that the overall periodic nonlinearity of this 1D encoder now is less than ± 10 pm.

2D HOMODYNE ENCODER

Principle

The principle of this novel 2D homodyne encoder is schematically shown in Figure 6. The encoder head consists of a beam splitting element and two modules which are mirror-symmetric with respect to the normal of the grating scale. If there are relative displacements Δx in X and Δz in Z direction between encoder head and grating scale, then $\Delta x - \Delta z$ and $\Delta x + \Delta z$ will be detected by module 1 and module 2 respectively. Therefore Δx and Δz can be measured simultaneously by this 2D encoder.

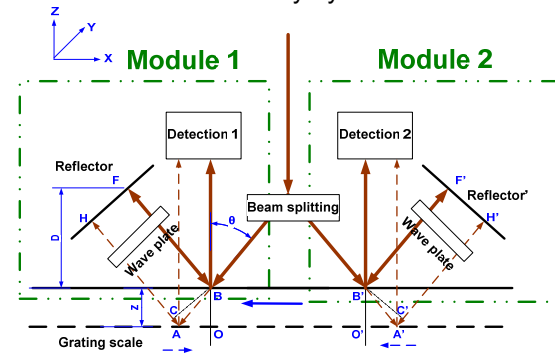


FIGURE 6. Schematic of the 2D homodyne encoder.

At module 1, incidence beam strikes the grating scale with an incidence angle θ , which is chosen to meet the grating equation: $p \sin(\theta) = \lambda$. In which, p is the period of the grating scale; λ is laser wavelength. After diffraction on the grating scale, the $+1^{\text{st}}$ diffraction order beam propagates along the normal of the grating scale; and the 0^{th} order is reflected back to the grating scale along the incoming path by the reflector and passes through a wave plate twice. Then the secondary -1^{st} diffraction order of the former 0^{th} diffraction order beam and the aforementioned $+1^{\text{st}}$ diffraction order beam are overlapped and interfere. Finally the phase information resulting

from the displacement of grating scale in X and Z directions is detected by detection 1. Then the Δx - Δz can be determined. Likewise, $\Delta x + \Delta z$ is determined by module 2.

Signal period of its X direction measurement is $p/2$. And that of Z direction measurement is $p \cdot \tan(2\theta) / (4 + 2 \cdot \tan(2\theta) \cdot \tan(\theta))$. The measurement range in Z direction is limited by the size of detectors.

Experiment

Up to now only the experiment of one module was performed as a preliminary proof of principle. The setup is shown in Figure 7. The period of grating scale and the wavelength of laser are the same as those of 1D encoder experiment.

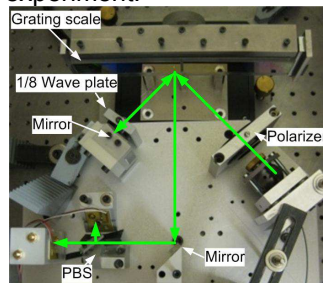


FIGURE 7. setup of the preliminary experiment of the novel 2D homodyne encoder.

During this experiment, piezostage moved in triangle waveform with amplitude of 10 μm at 0.1 Hz in X or Z direction. The signal periods of the 2D encoder are 416.67 nm in X direction and 345.58 nm in Z direction. The design measurement range in Z direction is ± 1.5 mm.

Since the main purpose of this preliminary experiment is only to prove the principle, a number of makeshift solutions have been adopted. Therefore quite high nonlinearities: ± 80 nm in X direction and ± 60 nm in Z direction were identified, as shown in Figure 8 which displays the residuals after linear fitting of the encoder readings.

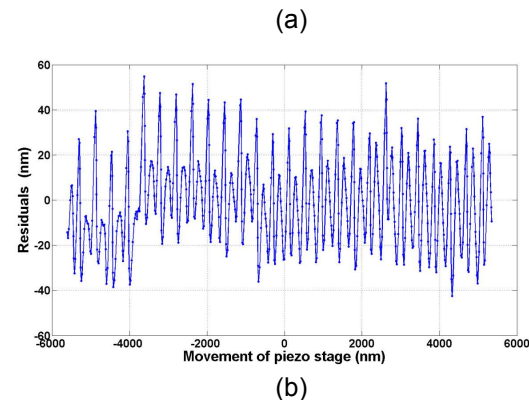
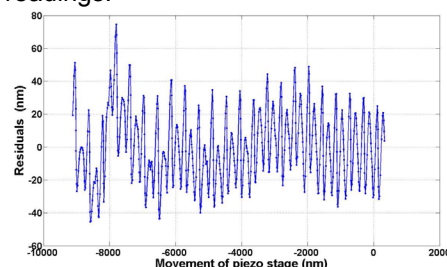


FIGURE 8. Nonlinearities of the 2D encoder: (a) in X direction; (b) in Z direction.

CONCLUSION

A 1D homodyne encoder is demonstrated, which has a periodic nonlinearity of ± 10 pm when a home built phase meter based on fast ADCs together with off-line Heydemann correction are employed. A novel 2D encoder is put forward and preliminarily proved by the proof-of-principle experiment.

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An automatic focusing module for White light scanning interferometer

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INSTRUCTIONS

Due to the drastic developments in the micro or nano techniques, the measurement of sub-micrometer dimensions has been an important issue. Many industrial inspection systems are employed to measure the precise dimensions of such micro- or nano-scale objects. During the conventional operation of inspection systems, users have firstly moved the scope's tube to the proper position at which the clear images is obtained in the field of view. Then, users have been able to start the inspection procedure to acquire the 2D imaging or 3D profiling. The human-operation could usually take so long time that it lowers the efficiency of the inspection. Thus, an automatic-focusing (AF) module must be introduced to improve the efficiency. Although many inspection systems now already have included the well-performed AF module, they have not been with good accuracy and efficiency for the white-light scanning interferometer (WSI). Due to the broadband light source, the low coherence length limited the range of fringe detection. Thus, it is difficult to obtain the focused image with clear fringes by most conventional AF modules.

The techniques of the auto-focusing are often divided into two types: the passive and active AF. The active type needs additional distance sensors, whereas the passive type only requires the image information and optical theory. Thus, the passive type costs less than the active type. In the expensive inspection systems, the passive AF method gradually becomes the best choice. Generally, the passive auto-focusing includes two parts: first, the focus indexes must be chosen and calculated according to an appropriate algorithm; second, the focus-searching technique is also utilized to efficiently determine the clear image with the highest focus index. In early years, some major functions for the AF of microscopes were compared to determine the best focus function [1]. Afterward,

Y. Sun, et al. [2] further organized the four categories with eighteen items for the AF algorithms. This paper developed an AF module for the WSI. This has a focus-searching technique with both the large-step searching for fast speed and small-step searching for high accuracy.

WHITE-LIGHT SCANNING INTERFEROMETER

Figure 1 presents the configuration of the Mirau-type WSI system [3]. The WSI system uses a broadband light source of low temporal coherence length. The light is incident from left to right onto the first beam splitter, where it is reflected downward onto the Mirau objective. The reflected beam is directed onto the reference mirror and the other onto the specimen. The beams reflected from the reference mirror and specimen are then interfered. Finally, the interfered beams pass through the first beam splitter onto the recording CCD. The CCD signals are then received by the image grabbing board installed in the computer, which provides functions of sensing, controlling, calculating, and etc. The CCD is equipped with an actuator to adjust its position to obtain proper imaging. The other actuator is installed on the objective to scan vertically to the surface of the specimen. Control of the actuators is also performed by the computer. Since the broadband source, white light, has a low coherence length, interference fringes exist only in regions in which the optical path difference (OPD) between the two beams is shorter than the coherence length of light source as predicted in Figure 2. The OPD changes as the precision actuator carries the Mirau objective in a vertical scanning direction. The surface profile of the specimen is obtained by analyzing the peak position of the coherence envelope of each measured pixel.

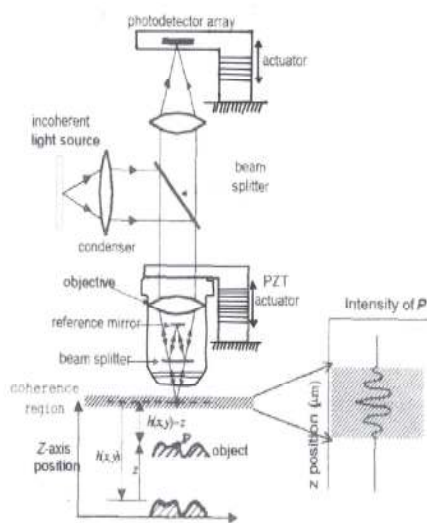


FIGURE 1. Schematic diagram of White-light Scanning Interferometer

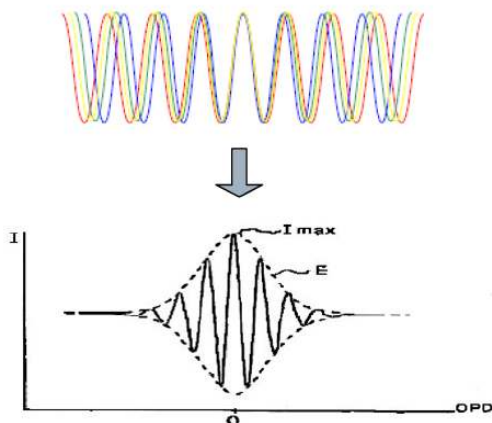


FIGURE 2. Interferogram produced by white light Interferometry

PRINCIPLE of AUTOMATIC FOCUSING

The scope tube with objective, beam splitters, optics, CCD and actuators lens moves to get the proper focus position. To avoid time consuming operation, frequent user intervention and operator-dependency, the AF module adopt a servo motor for its scope tube movement. In the modified design one extra beam splitter is placed in the optical path of the microscope. The beam splitter routes the reflected beam towards the specimen and the transmitted beam towards the CCD camera of the microscope. The position scope tube with the beam splitter is directly related to degree focusing because it changes brightness image which is used to calculate the degree. To obtain high degree of focusing and make sure of the focus position,

the beam splitter is placed exactly at equal distance from both the CCD and specimen. The overall mechanism of the AF in the WSI is shown in Figure 3. The CCD in WSI is used to obtain image which is then converted to the degree of focusing. A servo motor is applied as the vertical actuator of the scope tube so that the specimen can be vertically focused by controlling the servo. Every focusing motion, the focus index is calculated by the chosen focus algorithm until the highest index is achieved. This algorithm used Sobel operator to extract brightness of each pixel and then the degree of focusing was calculated from the brightness [4]. Moreover, to increase the speed of focusing, the servo receives the position command calculated from the focusing-searching technique to make the best movements. When the focus-searching technique determines the focused image, which has the maximum focus index, the servo stops focusing and the focusing algorithm finishes calculation. Then, the clear image with interfered fringes is observed in the field of view.

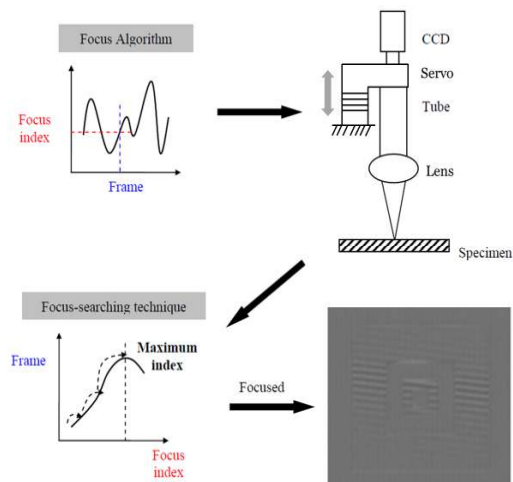


FIGURE 3. Proposed mechanism of the auto-focusing for WSI

ALGORITHM of AF MODULE

The AF module includes the object, CCD camera, optical lens platform, image grabbing board, computer, motion controller, drive unit and servo motor as shown in Figure 4. During the process of AF, it firstly estimates the direction and the distance of defocus based on the image analysis of the current frame and next frame image; then according to the search strategy, it adaptively adjusts the moving step and executes coarse searching procedure for adjustment in large searching range, or fine

searching procedure for adjustment in small searching range, until the best focus position is reached. The principle frame is shown in Figure 5, and the hill-climbing search is employed during the whole progress as shown in Figure 6.

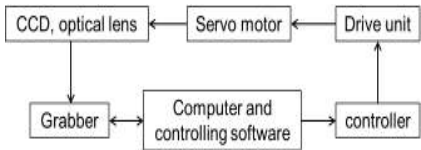


FIGURE 4. Block diagram of AF module

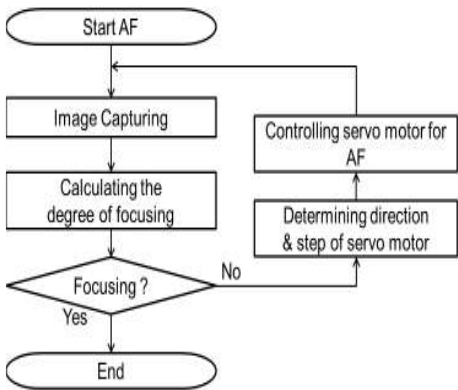


FIGURE 5. Algorithm of AF process

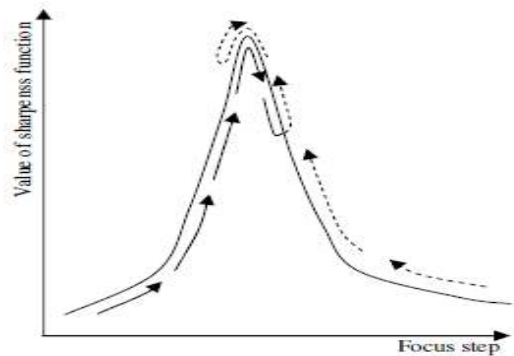


FIGURE 6. Hill-climbing search for AF

EXPERIMENT and DISCUSSION

The experiment used a program developed by Visual C++ which is shown in Figure 7. To verify the feasibility of AF module, a standard specimen is focused and measured. The standard specimen is certificated by Korea Research Institute of Standard and Science (KRISS). It has a pattern with step. Nominal value of the step height is 10 μm but the actual height is 9.65 μm . The height of the standard specimen is measured with the white-light scanning interferometer.

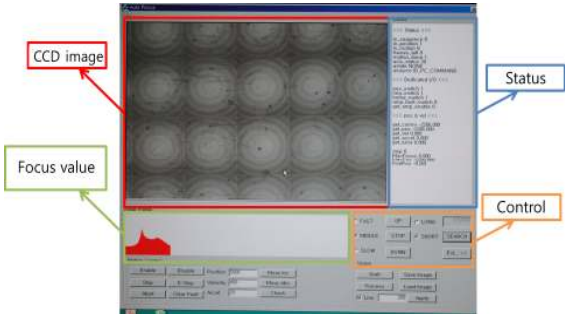


FIGURE 7. Control panel of AF module



FIGURE 8. WSI & Standard specimen

TABLE 1. Specifications of WSI

Components	Specifications
AMP	Model: ACURA 2000, INTEK PLUS Co.
CCD Camera	Model: TM200, PULNIX Co. Pixels: 768 (H) x 494 (V) Cell size: 8.4 μm x 9.8 μm
Lens	2.5X, NIKON Co.
PZT	Range: 100 μm , PI Co.

The experimental results shows that the system developed can not only adjust the focus position at high speed but also can measure the surface height of standard specimen within 10 second. Comparison of results with AF module and manual measurement facility is given table 2 and Figure 9 & 10.

TABLE 2. Comparison of results with AF module and manual setup

	AF module	Manual setup
Time	10 second	5~7 minute
Repeatability	Excellent (Ave. difference of 30 data taken at different time <5%)	Good (Ave. difference of 30 data taken at different time <10%)

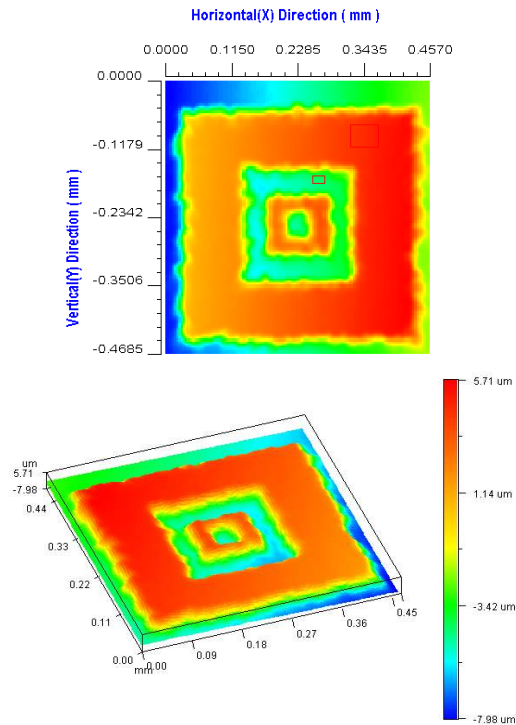


FIGURE 9. Measurement result with manual setup

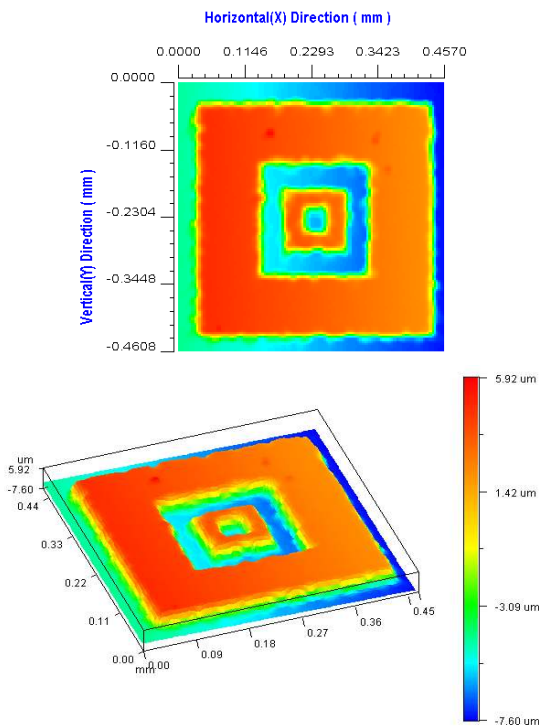


FIGURE 10. Measurement result with AF module

CONCLUSIONS

An automatic focusing module is developed in this study using Mirau's interferometer, beam splitters, CCD camera and a computer. The module can automatically adjust the focus position depending on how far it was initially. The surface profile and the heights between 30 different time for a standard specimen are measured using the developed system. The repeatability of the results shows that the module can detect the focus position within ten seconds and allow rapid acquisition of surface profile. Besides rapid detection of focus position automatically, low cost of the system components is also an added advantage of using this technique in comparison to other techniques.

ACKNOWLEDGEMENT

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Multi-Point Trilateration: A Novel Approach For Distributed Metrology

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INTRODUCTION

In this paper, a novel approach to distributed optical metrology is described and demonstrated that is based on optical trilateration. Trilateration is a method to determine the absolute 3D location of a target by measuring the distance to that target from four to five time-of-flight systems. Because the transceiver locations are known, the target coordinates are estimated from the intersection of the resulting spheres. The best known example of trilateration in use today is the global positioning system (GPS). By combining trilateration techniques with the high precision, distributed measurement capabilities of an FMCW chirped laser radar system, distributed metrology can be performed on an unprecedented amount of targets using only a handful of transceivers. Scaling of hundreds to thousands of targets per transceiver is possible.

Distributed metrology refers to the capability to accurately and efficiently reproduce length metrology across multiple targets. There are two critical reasons why distributed metrology remains a serious challenge for optical measurements with very high resolution. First, most optical metrology approaches, such as two or multi-color interferometers [1,2], can only make one distance measurement per transceiver. If multiple targets or stray reflections are observed by the transceiver, range pulling will introduce systematic errors that corrupt the measurement. This critical limitation has led to the necessity for large, complex optical Tx/Rx heads that require either spatial masking or the use of polarization to eliminate range-pulling and has been singled out as a critical challenge for these metrology techniques [3]. Second, because most optical metrology techniques can only make one distance measurement at a time, the number and complexity of optical transceivers becomes very large and cumbersome when the goal of the system is to make 3D coordinate estimates. For example, each target retroreflector (retro) requires at least three independent transceivers to measure the 3D coordinates. Each of these

transceivers must be critically aligned to the target retro, while maintaining high isolation between other retros on the structure of interest. Thus the brute scaling of multiple transceivers is an inelegant and impractical solution to the distributed metrology problem even for a fairly small number of targets.

Range-gated metrology is one possible solution to overcome this problem. The technique pairs a CW interferometer system (or an MSTAR system) with a coded waveform to overcome the single reflector limitation. In this distributed metrology approach, a more compact and scalable optical Tx/Rx head was also used. Demonstrations of this technique showed its effectiveness for stabilizing thermally induced length changes on an active optical truss [3]. The minimum range gate for this system is determined by the code bandwidth and a 0.6 m spacing was demonstrated. Because of electronic receiver and digitizer bandwidth limitations the minimum spacing is predicted to not decrease much below about 0.1 m.

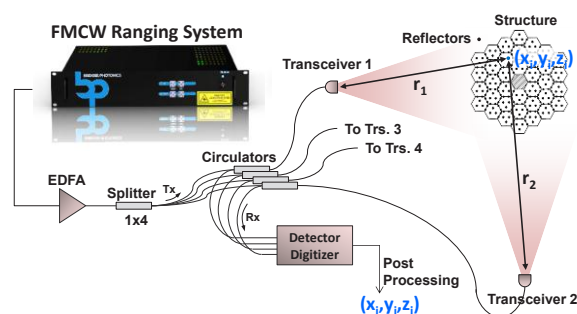


Figure 1. The multi-point trilateration system diagram for performing distributed metrology.

Laser radar systems have an inherent capability to provide distributed metrology as they are capable of measuring multiple targets within the transceiver field-of-view. Theoretically, laser radars are capable of making $N = \Delta R / R_W = c / 2BR_W$ resolvable target measurements. Here $\Delta R = c / 2B$ is the radar range resolution, c is the speed of light, B is the waveform bandwidth and R_W is the range window of the system. This is a dramatic

increase over traditional optical metrology approaches. In the past, laser radar was overlooked as a high resolution metrology tool, because the generation of large bandwidth optical waveforms was complex and because direct detect receivers were severely bandwidth constrained in a fashion similar to that of the range gated metrology system described above.

With the recent advent of actively stabilized swept laser sources used in combination with the FMCW laser radar technique, unrivaled optical bandwidths and range precisions have been measured [4]. Thus, laser radar systems have the potential to enable next generation distributed metrology with a significant improvement in the number of target measurements per transceiver.

SYSTEM DESIGN AND MODELING

Figure 1 shows the system diagram for a multi-point trilateration system based on an FMCW laser radar. The output from an extremely broadband actively stabilized swept laser source ($B \geq 4$ THz) is split and directed to four transceiver heads with large field of view. These heads (only two are shown) illuminate multiple retros across the structure of interest, which in this diagram is a deployable space structure such as a segmented primary mirror. In this diagram, the transceiver creates a reflected signal for a reference and both this reference and the return signals from the multiple retros are isolated using a circulator and detected. The retros are all resolved in a single laser radar range profile and measurements of their distances (r_1, r_2, r_3 and r_4) are used along with the knowledge of the transceiver locations to estimate the reflectors 3D coordinates.

There are three main constraints on the multi-point trilateration system. First, it is well known that trilateration requires large viewing angles between the transceivers in order to minimize uncertainty. Second, the laser radar for each of the transceivers will need to resolve and identify each of the retros within the range profile without any substantial overlap between peaks. While the problem becomes more complicated as the number of targets increases, the large range window and the fine range resolution of the FMCW laser radar system make the problem tractable. Finally, the transceiver must gather enough return light from each target in order to ensure that adequate precisions are obtained to meet the 3D coordinate estimation requirements.

There are many methodologies that can be used to determine placement of both the transceivers and the retros, but reliance upon a computational model and simulations is critical for appropriate retro placement. In our model, it was assumed that retros could be easily moved across the structure. A convenient transceiver geometry and a minimum peak separation distance were chosen. The minimum peak separation is required to be large enough so that range sidelobes from the peaks do not interfere with each other. Once these parameters are known, retros are initially populated at desirable locations. The expected ranges from all of the retros is then calculated and placed into a vector, r_i , for each transceiver, i . This vector is then sorted, a difference between adjacent elements is performed and distances smaller than the separation parameter are identified. Those retros must be moved to a new location. The next closest differences can also be used to identify in which direction the retro can be best moved. This movement needs to be performed serially for each transceiver as movement to separate retros in one transceiver's range profile can make the same retro closer to another retro in another transceiver's range profile. Care must also be taken for the expected range of motion expected from the structure.

To validate the algorithms and usefulness of this technique, the above retro placement methodology was used on a hexagonal mirror was performed. The structure is shown in Figure 2 along with an inset showing the placement of example retros. Once the final placement was determined, a full system model including photon budgets and noise sources in the laser ranging system was utilized to explore expected performance from the distributed metrology system.

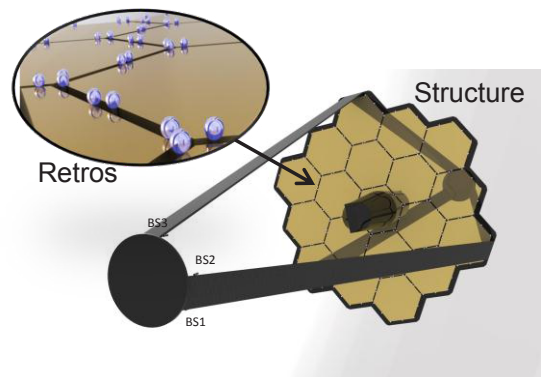


Figure 2. The simulated hexagonal mirror structure with example retro placement.

The retro positions were estimated from each transceiver and then a computationally efficient trilateration algorithm was used to estimate the 3D coordinates for each retro. Various deformations were applied across the structure and monitored using the simulated multi-point trilateration system. Figure 3 (left) shows an example for an actual parabolic error across the mirror surface. Figure 3 (right) is the result from the multi-point trilateration system. In this plot, the 3D coordinates are interpolated onto a finer grid and displayed as a colormap for visualization. Residual errors utilizing practical system parameters were below 200 nm. These results indicate the viability for the multi-point trilateration system.

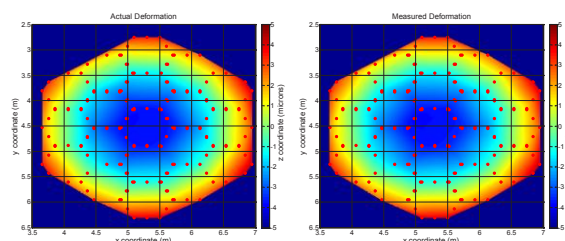


Figure 3. (Left) The simulated parabolic error in x and y with a 4 micron piston offset. Retro locations are shown as red data points. (Right) The estimated mirror shape from the system model showing very good agreement.

TWO DIMENSIONAL DEMONSTRATIONS

Two dimensional (2D) demonstrations were conducted as a first step towards a full system. The transceivers consisted of bare fibers with a mode field diameter of $11\text{ }\mu\text{m}$ allowing the transmit beams to expand to about a 20 cm diameter spot at a standoff distance of about 1 m. The transmit beams were subsequently overlapped to provide coverage on the retro target which was mounted on a 2D stage that was roughly aligned with the bisector between the transceivers (the y -coordinate). Before 2D trilateration could be performed, the locations of the transceivers needed to be determined relative to the stage.

This problem was actually similar in nature to the 2D trilateration problem. In this case, the unknowns are the transceiver coordinates rather than the position of the retro. Because the stage was well calibrated, a set of known positions were chosen for the target retro and measurements were performed simultaneously from each transceiver. The system of equations can be written as

$$\begin{aligned}(x - a_x)^2 + (y - a_y)^2 - r_a^2 &= \delta a^2 \\ (x - b_x)^2 + (y - b_y)^2 - r_b^2 &= \delta b^2\end{aligned}\quad (1)$$

Since the coordinates (x, y) for the retro are known and r_a and r_b were measured at each location, a nonlinear estimation was used to determine (a_x, a_y) and (b_x, b_y) , which are the coordinates of the transceivers.

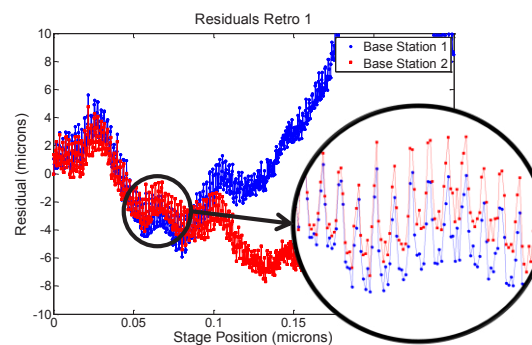


Figure 4. The residual errors from the two transceivers showing large systematic and periodic errors due to stage motion.

The positions of the transceivers were estimated to be $(0.5106, 1.1067)$ and $(0.5125, 1.1071)$. The residual errors from the transceiver coordinate estimation are shown in Figure 4. Systematic errors due to the stage motion including a 2 mm periodicity due to the lead screw are observed. The standard deviation is about $\pm 10\text{ }\mu\text{m}$.

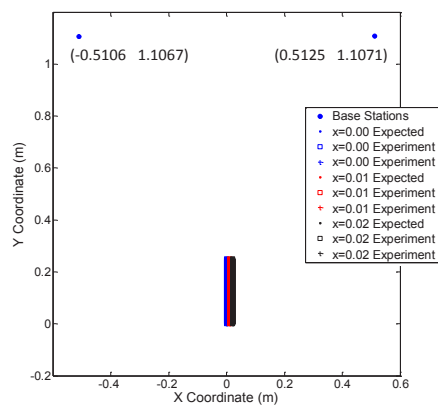


Figure 5. Results from the 2D trilateration tests.

Once the transceiver positions were estimated, 2D trilateration demonstrations were conducted. For these tests, the retro was moved in $250\text{ }\mu\text{m}$ increments in the positive x -direction for 25 cm. This was repeated twice for each y -position ($0, .01016\text{m}$ and $.02302\text{m}$) for a total of six experimental runs. Figure 5 shows the 2D trilateration results on a scale that shows the two

transceivers (blue dots) and the various estimated positions of the retro target. A zoom of the $y = 0.01016$ and 0.02302 m locations are shown in Figure 6. Note that the y scale covers 25 cm whereas the x dimension is about 20 microns. The runs were fairly repeatable with sub-3 micron standard deviations in either dimensions across all x positions.

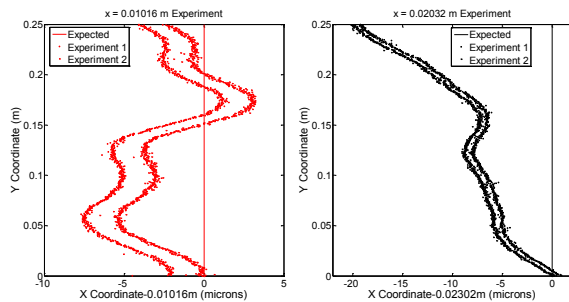


Figure 6. A zoom of the 2D trilateration results for $x = 0.01016$ and 0.02032 m. The errors are primarily due to the repeatability of the stages as the precision of the measurement system is around 200 nm.

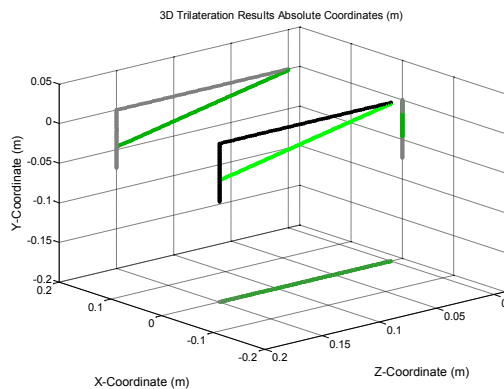


Figure 7. The measured coordinates for the retro when the stage was moved by 1 mm in the z -direction and 300 mm in the y -direction steps and their projections into the respective planes.

THREE DIMENSIONAL DEMONSTRATIONS

The system was extended into three dimensions by adding a third transceiver and by adding an additional dimension to the stage. Similar procedures were followed to determine the transceiver locations. The 3D trilateration results of two runs can be seen in Figure 7. One run of the retro moving in the z -direction (toward the transceivers) and then the y -direction (down) and another run moving simultaneously in both the y - and z -directions. The coordinates are then projected on the yz on the left, xy to the right and xz at the bottom to show the motion.

The precision was found to be 1.3, 3.9 and 0.9 microns respectively for x , y and z respectively.

MULTIPOINT TRILATERATION

Finally, five additional retro targets were added to the 3D stage to test the multi-point trilateration capability. The range peaks from each transceiver's range profile were identified for each retro and the center locations were measured. This resulted in a total of eighteen peak centroid measurements (three transceivers times six retos). By using those eighteen peak centroid measurements, the five moving retro coordinates are determined. The estimated 3D coordinates are shown in Figure 8 for motion in the y -dimension. A precision of 1 micron was measured for the y -motion.

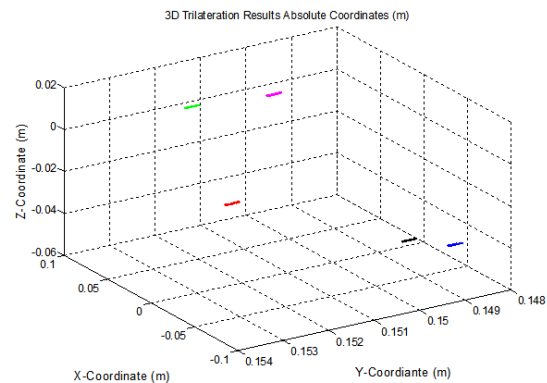


Figure 8. The 3D coordinates of the retos when the stage was stepped by 20 microns in the z -direction for 16 steps.

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SENSITIVITY OF OPTICAL FIBERS TO MECHANICAL AND THERMAL PERTURBATIONS FOR INTERFEROMETRIC APPLICATIONS

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INTRODUCTION

Heterodyne displacement interferometers have been used in many different metrology applications. Specific uses include stage calibration and feedback systems, laser alignment [1], and gravity wave detection [2]. Currently, most interferometers employ collinear, orthogonally polarized heterodyne beams that are delivered via free space steering mirrors. As more systems use displacement interferometers, there is a trend to use a single heterodyne laser source for many axes, such as in lithography stage feedbacks systems. This presents a coupled alignment problem for each interferometer axis, as depicted in Figure 1, because drift and misalignment of steering components affect each interferometer differently [3].

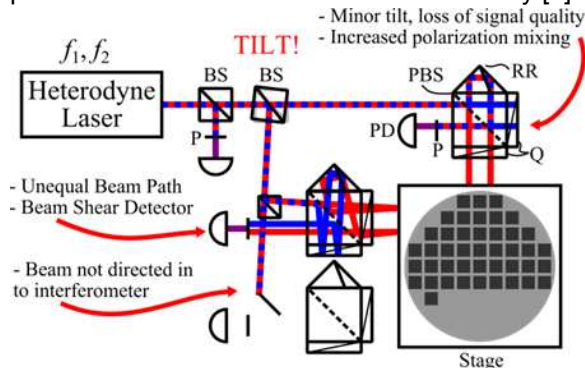


FIGURE 1. Example of how a coupled alignment via beam steering optics can adversely affect multiple interferometers, leading to needing a complete system alignment that can be costly and time consuming.

For these applications, it is advantageous to have each individual interferometer fiber-delivered, decoupling alignment between each interferometer axis. This also has the added benefit of potentially reducing the size of the metrology frame and loop in the system because beam steering components are eliminated.

There are many advantages to using a fiber-delivered interferometer system, including relocation of heat sources (laser and electronics) far away from the system, decoupling interferometer alignment from source alignment, and remote sensing where the interferometer may need to be repositioned, such as in optical probing. However, fiber delivery introduces new error sources that can lead to uncertainty in the displacement measurement [4]. These include polarization mixing, reduced beam wavefront quality, external perturbation-dependent power fluctuations, and Doppler shifts induced into the interferometer due to fiber physical and optical length changes from thermal fluctuations and stress.

The effects of temperature and pressure on the optical path of a single-mode fiber have been calculated and experimentally demonstrated [5]. The effects of stress have also been investigated both theoretically and experimentally [6].

The standard form of the ideal signal detected by a displacement interferometer is given by the equation

$$I_m(t) \propto A \cos(\omega_s t - \Delta\phi), \quad (1)$$

where A is the dot product of the vectors of the measurement and reference electric fields, ω_s is the angular split frequency in radians/second, and $\Delta\phi$ is the displacement-dependent phase shift in radians [7]. This measured signal is then typically multiplied by the in-phase and quadrature reference signals,

$$I_{r,i}(t) \propto A \cos(\omega_s t) \quad \text{and} \quad (2)$$

$$I_{r,q}(t) \propto A \sin(\omega_s t), \quad (3)$$

and low-pass filtered to remove the $2\omega_s$ component. This generates the in-phase component (I) and quadrature component (Q). The displacement-dependent phase is then computed

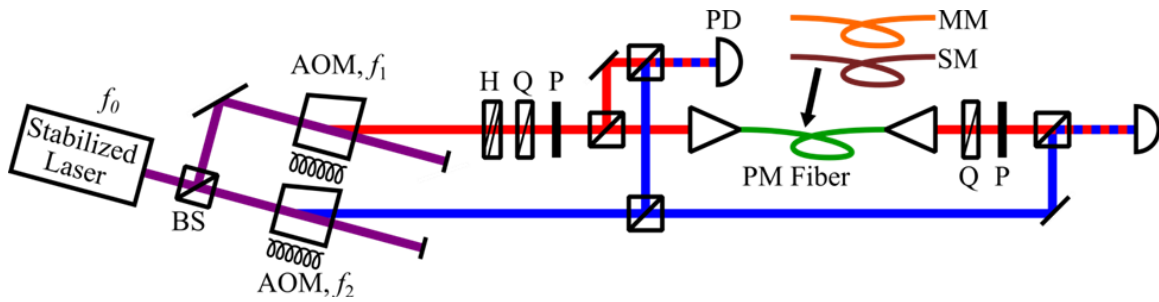


FIGURE 2. Setup for quantifying the consistency of the polarization angle, ellipticity, output power, and Doppler shift for different fibers. The goal is to determine the conditions and components that are needed for fiber launching and outcoupling collimating to ensure a known polarization state and Doppler shift can be maintained despite external perturbations on the fiber.

using

$$\Delta\phi = \arctan\left(\frac{Q}{I}\right). \quad (4)$$

We also determined the R value by adding I and Q in quadrature. The R value is proportional to the square of A and, assuming that the power from each arm of the interferometer is constant, proportional to the contrast ratio.

Fibers cause a greater phase shift than free-space propagation, but the phase shifts caused by the fiber delivery affect both the measurement and reference signal equally, if the interferometer is designed for this. This simultaneous phase shift merely has the effect of putting a time-dependence on ω_s without affecting $\Delta\phi$ - the displacement-induced phase shift. However, finite sample rates limit the magnitude of those common-path Doppler shifts to ensure no aliasing or unwrapping problems.

In this proceedings, we consider the three most common types of optical fiber: multimode, single mode and polarization maintaining. Multimode fiber is ideal for launching since it has a much larger core (65+ μm) area. In multimode fiber, the detected phase is dependent on the power weighted average of the accrued phase of all excited modes. Unfortunately, the relative power of the modes (and thus the weighting of the average) is affected by bends [8] and stress [9]. Single mode fiber is not as affected by mode mixing, since only one mode is guided, though it is more difficult to launch than multimode fiber due to a much smaller core area (5 μm). Unfortunately, both inherent and perturbation-induced birefringence in the fiber can reduce the signal as men-

tioned earlier. Polarization-maintaining (PM) fiber exists specifically to mitigate this problem. PM fiber has large birefringence built in to ensure that most perturbations to the polarization will result in light coupled to a polarization in which the light is not in-phase with itself. However, PM fiber is also the least desirable for launching, since the core is smaller, and the azimuthal alignment at launch is much more important due to the high birefringence creating specific axes along which the light must be launched.

EXPERIMENTAL PROCEDURES

In this Proceedings, we have tested the three fibers discussed in the previous section, using the setup shown in Figure 2. We specifically used ThorLabs P1-630A-FC-2 single mode fiber, ThorLabs P1-630PM-FC-2 PM fiber, and ThorLabs M31L02 graded index (GRIN) multimode fiber. A beam from a single mode stabilized HeNe laser was be separated into two different paths, where each beam was frequency up-shifted using separate acousto-optic modulators (AOMs) driven at slightly different RF frequencies, resulting in a constant, known heterodyne frequency of 70 kHz or 5 MHz, as explained later.

We measured the R value using a commercial lock-in amplifier, which limited our split frequency to 70 kHz and our sampling frequency to 20 kSPS for those tests. These limits caused unwrapping problems for the phase because of the high Doppler shifts induced, so we were unable to use it to measure the phase. To process the high dynamic phase signals, the 5 MHz heterodyne frequency was used.

The phasemeter we used was built in an FPGA demonstration board containing two high-speed ADC channels. The analog reference and measurement signals are converted to 14-bit digital signals with a 50 MSPS sample rate by the high-speed ADC board for signal processing in FPGA. Then an in-phase and quadrature algorithm written in Verilog/VHDL runs in realtime on the FPGA to get the phase shift between the two 5 MHz signals. The data that contains phase shift information are stored in on-board SDRAM. After measurement, software on a PC reads the data out of memory for further processing. Currently, we can demonstrate wire tests achieving a noise floor of 0.4 mrad (20 pm). In future research, we will design a more specific and integrative PCB and choose a more efficient interface for data logging.

We tested a number of situations to produce 'typical' perturbations encountered in real-world metrology situations as well as applying no perturbations to investigate the general stability of each fiber to environmental fluctuations in a laboratory. The three perturbations we applied were lightly tapping the fiber with a screwdriver to simulate something bumping into the fiber (tapping), applying heat by breathing on it (thermal), and gently moving the fiber to simulate the interferometer moving on a stacked stage (move). These three perturbations should provide an accurate representation of the suitability of the fibers under test. The experimental setup was not isolated from normal laboratory environmental fluctuations during these tests.

RESULTS

The maximum Doppler shift recorded with no perturbations intentionally applied to the fiber is shown in Table 1. The standard deviation of the Doppler shift with no perturbations applied is near 11.5 Hz for each fiber. This noise level is the same for all three fibers. Additionally, there are no significant differences to either the peak or standard deviation (approximately 12 Hz for all fibers) of the noise level when applying heat to the fiber as shown in Table 1.

The recorded phase while lightly tapping on the fibers with a screwdriver is in Figure 3. The instantaneous Doppler shift for one characteristic event is shown in Figure 4 and the maximum Doppler shift is shown in Table 1. The PM fiber has a significantly larger peak Doppler shift than the other fiber types, which had very similar maxima. Figure 5 shows that the R -value of each fiber

TABLE 1. The peak Doppler shift in each case is shown in this table. The columns denote different fibers and the rows denote different perturbations.

Perturbation	MM	PM	SM
None	159 Hz	120 Hz	117 Hz
Thermal	131 Hz	113 Hz	109 Hz
Move	8.3 kHz	2.1 kHz	5.4 kHz
Tapping	13 kHz	24 kHz	13 kHz

is similarly affected by tapping.

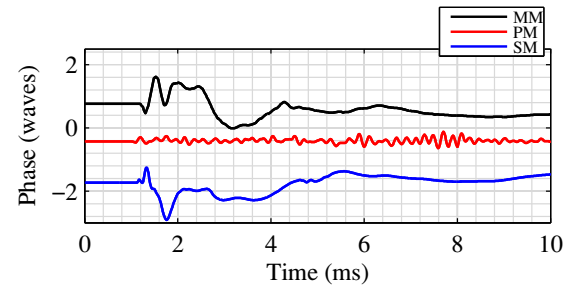


FIGURE 3. The phase over time at the beginning of one tap.

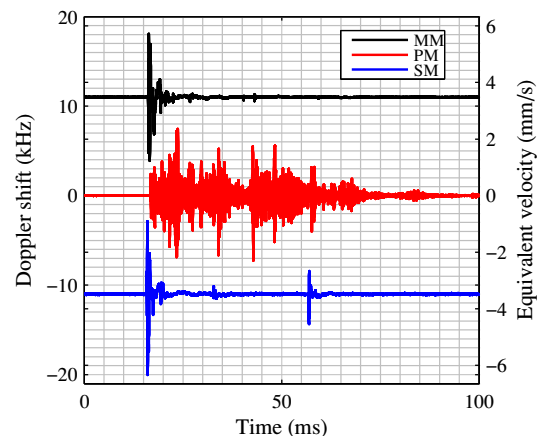


FIGURE 4. The instantaneous Doppler shift of one tap as well as the velocity that is equivalent to those Doppler shifts. The time of each tap has been offset for clarity.

The peak Doppler shift that results from moving the fiber slowly is shown in Table 1. In this test, the PM fiber results in the smallest peak Doppler shift, with the MM fiber resulting in the highest peak Doppler shift. However, there is a clear difference when considering the R -values of the three fibers, as shown in Figure 5. The change in the R -value using the multimode fiber is far larger than that of the single mode fiber, whereas the

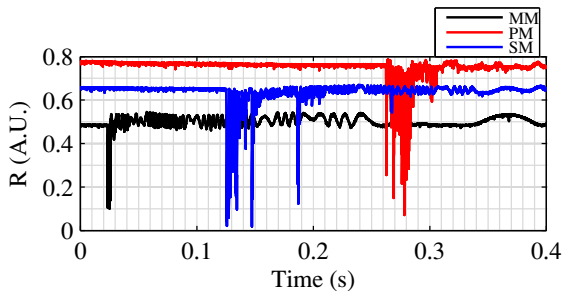


FIGURE 5. The R value using the fibers during one tap in arbitrary units with slightly different scalings and to provide offset for clarity. Additionally, the timing of each tap has been offset.

R -value using the PM fiber is almost unchanged.

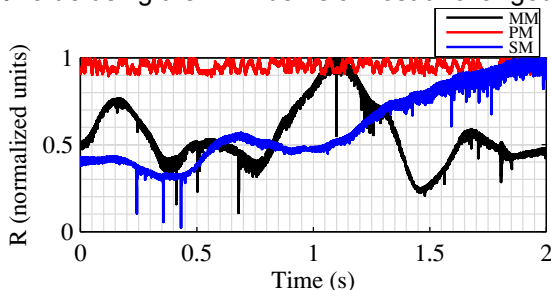


FIGURE 6. The R values using the fibers normalized to the same maximum to show the change.

CONCLUSIONS AND FUTURE WORK

The advantage of PM fiber is clear based on these results. The Doppler shift of all three is similar under most conditions. However, when being moved, using PM fiber resulted in the lowest Doppler shift, and the most stable contrast ratio. The single mode fiber results in a significant Doppler shift, but the contrast ratio is more stable than the multimode fiber.

Future work includes determining the contrast ratio directly, rather than the R -value. Knowing this contrast ratio would allow us to determine whether the contrast ratio remains within an acceptable range, or whether it is reduced too much to allow precise measurements. It is also important to know the amount of ellipticity in the output of the fiber, because it has already been established that non-ideal polarizations can result in periodic non-linearity [10]. These measurements can be accomplished using a polarimeter. Finally, we will also investigate ways to improve both sin-

gle mode and multimode fiber for displacement interferometry applications.

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Optomechanical characterization of large wafer stepper-optics with respect to centering errors, lens distances and center thicknesses

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One key technology in mass microchip-production is an optical projection system (wafer stepper) for deep UV photolithography. Modern wafer stepper-optics featuring sub-200nm resolution, consist of 20 to 25 glass elements with a weight of up to 1000 kg.

After the careful glass element-manufacturing, the optical performance of the system is highly influenced by the precise mechanical alignment of all individual lens elements. The imaging quality of such systems is extremely sensitive to centering errors as they may severely degrade the imaging quality. Therefore, the optical design of a stepper optic is tightly toleranced for the alignment in the sub- μm and arc second range.



FIGURE 1: Measurement instrument for the optomechanical characterization of large optical systems, e.g. stepper objective-lenses with control cabinet and PC.

Measurement instruments for real time observation during assembling and final characterization with a high resolution are rare for these types of massive optics and ultimate engineering requirements. Techniques like interferometry are able to determine the wave front, but the meas-

urement gives no quantitative results for the alignment errors of all single elements.

We will present the first commercially available high-resolution optical measurement system, which is specifically designed for large optics. For the complete optomechanical characterization, the instrument combines two high resolution focusing autocollimators and a low coherence-interferometer. The autocollimators reveal the exact centering errors of the optical elements and the low coherence interferometer determines the distances between surfaces. Both methods are non-destructive, quantitative and can be applied to assembled multi-element optics [1,2]. That means geometrical positions of all single surfaces of the final optical system can be precisely analyzed. Beside the final optomechanical characterization of the complete lens assembly, the instrument is suitable for precise lens element centering during the assembly process and automated lens-to-mount alignment.

The system performs quantitative non-contact measurements of centering errors of single elements and fully assembled optics with a measurement accuracy of $0.1 \mu\text{m}$, and accurate determination of air gap-distances and lens center-thicknesses with a measurement accuracy of $1.0 \mu\text{m}$.

TABLE 1: Overview of the instrument accuracy and maximum sample properties

Centering Error Accuracy	$0.1 \mu\text{m}$
Lens thickness Accuracy	$1 \mu\text{m}$
Air gap Accuracy	$1 \mu\text{m}$
Maximum Sample Height	2 m
Maximum Sample Weight	1500 kg
Maximum Sample Diameter	800 mm

Figure 1 shows a photograph of the measurement system (instrument stand, the control cabinet and the PC table) and Table 1 gives an overview of the main instrument specifications.

The fully PC-controlled instrument consists of a granite table with a linear air-bearing granite-column. This all-granite, all-air-bearing construction provides exceptional stiffness and thermal stability. The resulting extremely smooth motion enables the unrivaled accuracy (Tab.1).

The maximum sample diameter, given by the diameter of the rotation air bearing, is 800 mm with a maximum load of 1500 kg. To load such a heavy sample onto the instrument the upper measurement head on the linear air-bearing can be pivoted away to clear the room.

Measurement of the centering errors

The measurement of the centering error is carried out with an electronic autocollimator. For every optical surface of a system under test the position of the autocollimator and a suitable focusing optic are chosen in such way that the light emerging from the autocollimator is projected almost perpendicularly onto the surface. This results when the absolute position of the focus position along the optical axis coincides with the z-position of the center of curvature of the surface. A fraction of the light will be back reflected and forms an autocollimation image of the autocollimator target (usually a cross hair) on a digital camera device, e.g., a CCD camera. If the center of curvature of a surface is decentered with respect to a given reference axis of the measurement system, the image of the cross will be shifted. This can precisely be quantified when rotating the sample by the high precision air bearing. In this case the path of the cross hair image forms a circle on the CCD camera. The center of the circle is determined and diameter of the circle is related to the centering error of the surface (see Fig. 2) [1]. The last point of the circle yields the direction of the centering error.

For the measurement of the centers of curvature of inner optical surfaces within an assembled system the principal concept stays the same. But in order to provide an almost perpendicular incidence of the measurement light coming from the autocollimator onto the surface under test, the refraction of the optics that were traveled through to reach the surface need to be considered. The axial position of the center of curvature of such surfaces can easily be calculated if the basic design parameters of all surfaces are known: the radius of curvature, refractive index and the lens thickness. Furthermore, these pa-

rameters have to be taken into account for calculating the positions of the corresponding centers of curvature [3].

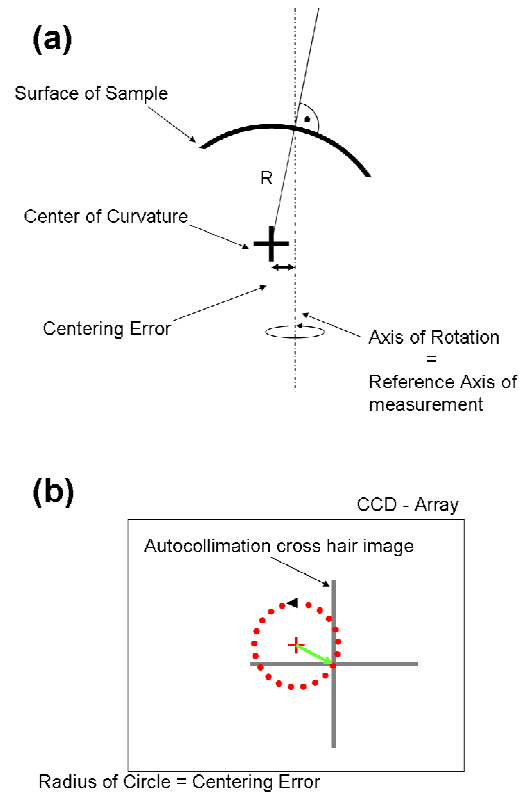


Figure 2: Measurement of the xy-position of the center of curvature with a focusing autocollimator: (a) Centering Error of the upper center of curvature to the axis of rotation, (b) circular movement of the autocollimation cross hair-image on the CCD Array.

In the measurement the surfaces are successively analyzed for the deviation of each center of curvature with respect to the reference axis. In the presented measurement system, the rotation axis of the air bearing represents the reference of each measurement. The obtained data are processed in order to provide the shift and tilt of an individual lens or group of lenses. In the analysis, the centering errors can also be transferred with respect to a different axis, e.g., the best-fit axis through all centers of curvature. Alternatively, it can be the optical axis of a single element of a defined group of lenses. In addition, the determination of the orientation of the best-fit axis itself represents a robust indicator for the alignment of the sample with respect to the reference axis.

The operator obtains a numerical and graphical output of the complete centering status regarding shift and tilt of all surfaces. The results can be transferred to a different reference system.

Measurement of air gaps and lens center thicknesses

The measurement of air gaps and lens thicknesses yields a different set of parameters for the evaluation of mechanical errors. Lens gaps and center thicknesses are defined as the geometrical distance between neighboring optical surfaces along the optical axis of an optical system. In this contribution, the positions of the optical surfaces are determined by means of low coherent interferometry.

In general, low coherent interferometry is based on the determination of the position of zero group delay difference within an interferometer system. In an interferometer, e.g., a classical Michelson-type (Fig. 3), the light coming from a low coherent light source, e.g., a spectrally broadened light source is split by a beam splitter into an object and a reference wave. In the presented case, a superluminescent diode (SLD) at 1310 nm with a coherence length of approx. 25 μm was used. The object wave illuminates the lens system along its optical axis. In every surface, a fraction of the incoming light beam is reflected. This light is guided to a photo detector where it is superimposed with the reference light that is varied in the optical path length (OPL) by an optical delay line. When analyzing the resulting intensity as a function of the delay (Fig. 3), interference patterns are observed whenever the optical path lengths in the two interferometer arms coincide. There, the group delay difference equals zero. Thus, the location of every object surface can be determined from the positions of the maximal modulation in the interference pattern.

An important prerequisite for the detection of every single optical surface is a sufficient intensity of the back-reflected light. Besides the optimized focusing of the object beam to obtain reflections with balanced intensities from all surfaces, it is essential to align an accurate coincidence of the measurement axis of the low coherent interferometer and the optical axis of the sample. This is a critical issue for optics with spherical surfaces, and of particular importance for multi-surface samples.

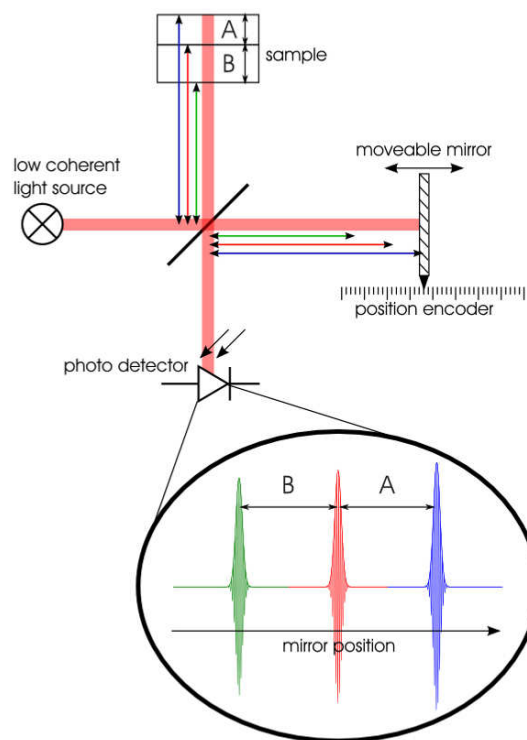


Figure 3: Measurement principle for the determination of air gaps and center thicknesses

The particular advantage of the presented measurement system is the combination of the low coherent method together with techniques for the determination of centering errors.

The centering error measurement is able to determine and align the lens group-axis of interest to the measurement axis of the interferometer. The group-axis of interest is for example the best-fit axis through all centers of curvatures, which is identical to the optical axis of the lens assembly if there is no centering error.

The measurement head of the low coherent interferometer is part of the dual measurement head carried by the linear air-bearing granite column (see Fig. 1). The beam is in coincidence to rotation axis of the rotation air bearing and with the linear air bearing it is stable in its adjustment.

Therefore the orientation of the low coherent interferometer-axis is well known and the sample axis can be aligned for an optimized measurement. These conditions ensure that exactly the lens center-thicknesses are measured by the low-coherence interferometer [2].

Live centering of elements

The observation of the centering state with the autocollimator is suitable and typically used for live position-tracking and fast element alignment.

If the cross image from the autocollimator is focused to the upper center of curvature, the position of the cross hair on the CCD camera gives the real position of the surface.

During the lens stacking, the rotation axis of the air bearing is an excellent reference for the assembly as it is stable in time, at all heights and for all lens elements.

As described before, the rotation axis is automatically determined and displayed after one revolution. Therefore, the procedure is to put the lens on top of the previously centered lenses, focus into the center of curvature of the upper surface, determine the rotation axis by one revolution and center the lens.

After the centering the low-coherence interferometer easily reveals the air gap between the upper and the second lens, because all lens elements are centered to the rotation axis which is the measurement axis of the low coherence interferometer. This method makes the lens stacking easier, faster and at the same time very precise.

Automated lens-to-mount alignment

An additional option is to use the centering error-measurement to align lenses with respect to a mechanical reference, e.g. a mount. Figure 4 shows a photograph of an automatic setup for the exact lens-to-mount centering. The setup is equipped with mechanical or optical distance sensors and three high-precision actuators mounted on a separate vertical stage to access and align every single lens element inside of the assembly. The full travel of this stage is 1500 mm.

The distance sensors are used to determine the orientation of the mount tube in space during one revolution. For every lens element, the displacement between the center of curvature and the location where the calculated housing penetrates the image plane (of the center of curvature image) is determined.

For the automated alignment of the lens the stage three actuators push the lens precisely to this point in order to ensure a minimized center-

ing error with respect to the mechanical axis of the mount. This positioning procedure of the lens can be live monitored during the alignment. The actuators are stepper motors or piezo actuators with a force of 600 Newton per actuator. Additionally, this bonding unit is equipped with a dispenser for glue and UV light sources.

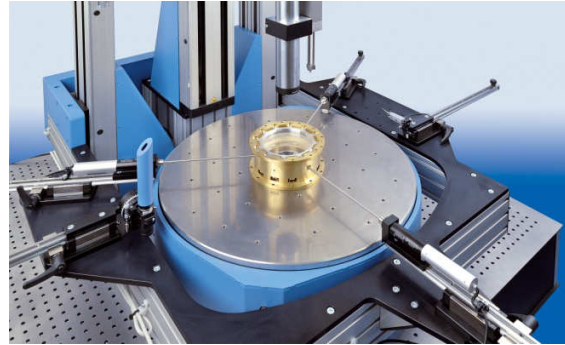


FIGURE 4: *Precise lens to mount centering with three automatic actuators.*

In summary, the presented instrument concentrates unique high resolution measurement instruments and procedures that allow the user the precise, fast and fully automated assembly of large wafer steppers-optics and a very accurate characterization of the whole lens.

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PRECISION FLEXURE STAGE UTILIZING PARASITIC MOTION

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INTRODUCTION

The requirement for frequency stabilized laser sources has driven the development of many different laser sources. Iodine stabilized lasers [1], Zeeman lasers [2], and two mode intensity balanced lasers [3] have been developed for precision systems requiring a stable frequency source. In these systems, frequency stability typically takes precedence over output power. To increase output power, while retaining high frequency stability, a three mode gain medium stabilized laser was proposed and has shown a fractional frequency stability ($\Delta f/f$) of better than one part in 10^{10} , [4, 5]. This was achieved by detecting an intrinsic mixed mode signal and thermally controlling the optical resonator cavity length, thus stabilizing the absolute frequency. In order to achieve higher levels of frequency stability it may be necessary to decouple the resonator length changes from the thermal fluctuations. To accomplish this, we propose a mechanically coupled laser resonator mirror, and a separate temperature controller to stabilize the tube temperature. We present here a flexure motion stage with a range of 200 nm, and a resolution of 20 pm for controlling the motion of a cavity mirror to stabilize the output frequency of a HeNe gas laser.

DESIGN REQUIREMENTS

Maximizing output power and frequency stability presents a challenge in the design of a new three mode stabilized laser system. First, the optical cavity should be maximized while maintaining at most three modes. For the 1.5 GHz HeNe gain spectrum, this translates to a free spectral range of 375 MHz or a cavity length of about 400 mm. This should ensure at least 4 mW of optical power is available from the central mode for use in the intended application. In order to increase the frequency stability to near Iodine stabilized lasers levels, the frequency stability must be better than 2.5 parts in 10^{11} [6]. To achieve this level of frequency stability, it is necessary to control the optical resonator length in this new HeNe gain medium stabilized laser to within 4 pm. This

work focuses on the design of a flexure stage for mounting one cavity mirror using parasitic motion to translate nanometer-level motion to parasitic motion on the picometer scale. This then allows for decoupling the mechanical motion and thermal properties of the laser cavity.

PARASITIC FLEXURE MOTION

A simple, compact mirror stage must be capable of traveling nanometers while maintaining picometer resolution. One method to accomplish this is to use the parasitic motion of a flexure stage. Using Euler-Bernoulli beam theory [7], the end deformation of a beam can be readily and repeatedly determined. This can be illustrated with the two flexure system as seen in Figure 1. The

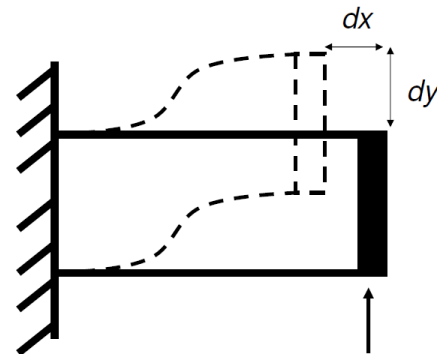


FIGURE 1. This figure shows the motion dynamics of the flexure stage under a force applied in the y direction, perpendicular to the beam axis. The resulting motions dy and dx are the piezo actuation and the parasitic motion, respectively.

two flexures ensure that the front face of the flexure system remains perpendicular to the beam axis (dx direction), while allowing a degree of freedom perpendicular to the beam axis (dy direction). This perpendicular motion is not the primary interest in this research, rather the parasitic motion is most important, as it allows a small degree of motion with high resolution by effectively leveraging the motion of the stage in the actuation direction. As a prescribed force (or distance) is enforced upon the flexure system, geometrical re-

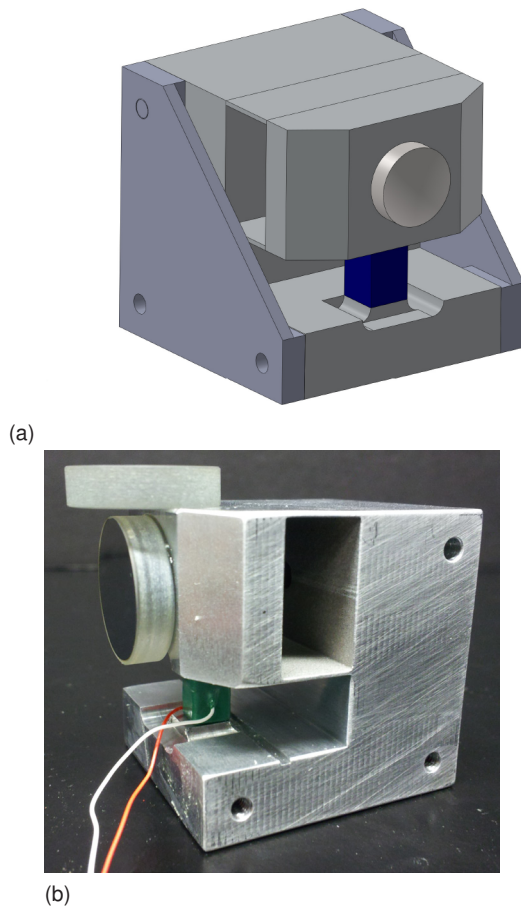


FIGURE 2. a.) Conceptual rendering of the flexure stage with the piezo actuator installed and side braces attached. b.) A picture of the actual stage, with mirrors attached to both axes of motion, without side braces.

lations and linear approximations can be used to estimate the parasitic motion. A solid model of the stage is shown in Figure 2a, and the actual flexure stage can be seen in Figure 2b. The piezoelectric actuator can output approximately $9.1\text{ }\mu\text{m}$ of motion. Based on the reduction ratio estimated with a linear approximation, this should be should reduce to 85 nm with a resolution of 20 pm . This stage's first resonance is at 7.1 kHz and has overall dimensions of 40 mm by 40 mm by 41 mm .

MODELING OF MOTION

Various modeling methods for this flexure stage have been investigated, including linear approximations, elasticity theory (i.e. Airy Stress Function), and Finite Element Analysis (FEA). Euler-Bernoulli beam theory and simple geometry, as seen in Figure 3, allows for estimating the parasitic motion. The predicted parasitic motion is

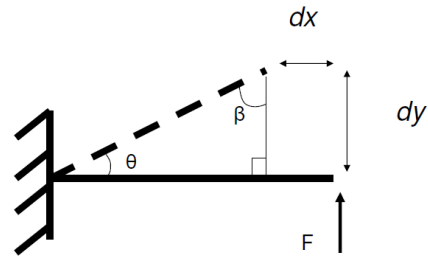


FIGURE 3. A linear approximation including parasitic (dx) and on axis (dy in the direction of the piezo) motion. θ and β can be determined if the length of the beam is assumed to be constant.

estimated to be 85 nm for a piezoelectric actuator displacement of $9.1\text{ }\mu\text{m}$. The linear approximation agrees closely with an elasticity model for motion perpendicular to the axes of the flexures, although the parasitic motion is not easily determined due to certain boundary conditions applied. Finally, using FEA has not proven fruitful for determining the parasitic motion for this system. Solutions returned from Solidworks' built-in FEA program have returned inappropriately large estimates of the parasitic motion; for a displacement of $9.1\text{ }\mu\text{m}$ perpendicular to the axis of the beam, the parasitic motion returned was $3\text{ }\mu\text{m}$, which is unrealistic given the geometry of the system.

EXPERIMENTAL SETUP

A completed stage, manufactured by OptiPro Systems, was mounted in the measurement arm of a compact, fiber delivered displacement interferometer [8]. The stage was first oriented to measure the displacement in the 'y' direction to measure and calibrate the motion of the piezo, after which the 'x' axis was measured. A SR830 Lock-In Amplifier was used to measure the phase from the interferometer. An xPC Target computer with a NI PCI-6052E data acquisition card created a real time PID controller and data recording tool. Figure 4 shows a photograph of the setup.

PRELIMINARY RESULTS

The goal for initial measurements of this stage was to characterize the relationship between the applied voltage and the piezo and parasitic stage motion. Measurements of the stage show that the relationship between displacement and applied voltage to the piezo is approximately linear for both axes of motion. The stage was displaced using a linear voltage ramp as seen in Figure 5;

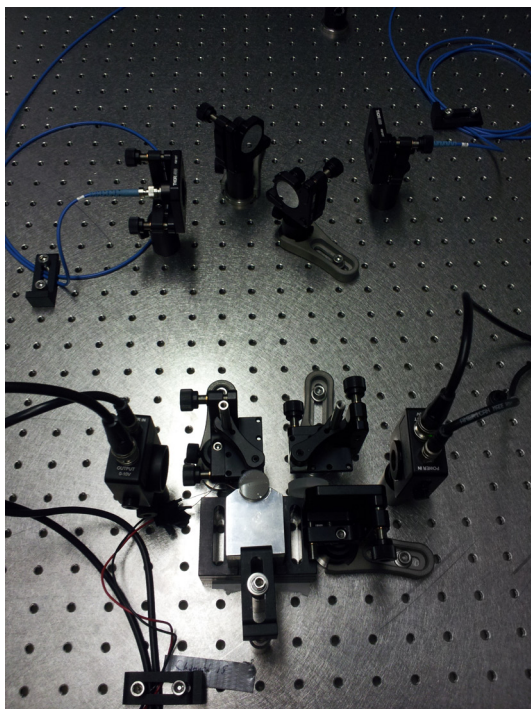


FIGURE 4. A photograph of the heterodyne interferometer setup. The measurement and reference arms are fiber coupled from the AOM outputs. The xPC Target PC is connected to the photo detectors, as well as the piezo amplifier for a driving signal.

these results were averaged 10 times before analysis. A polynomial can then be fit to this measured data, as shown in Figure 6b and Figure 6b. The polynomial fitting allows creating an accurate model to convert applied voltage to linear displacement. By doing so, the main motion axis can be controlled to nanometers, enabling picometer level parasitic motion for controlling the laser cavity mirror. The piezo's behavior is slightly nonlinear, though this will not play a large role in the operation of the stage, because a nominal voltage (e.g. 70 V) will be applied, and the stage actuated from this nominal position. Operating in this manner ensures that most of the motion occurs in a voltage range that is more linear than if operating only with low voltages (<10 V). To show the robustness of the current control system, various input signals were used. One example is a simple sine wave, as shown in Figure 7, where both the input signal and the measured displacement for the piezo motion of the stage are plotted. Overshoot can be seen at the start of the sine wave, though it disappears after the second period. To ensure that the control system is accurate down

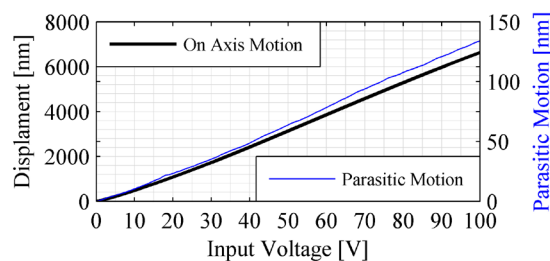
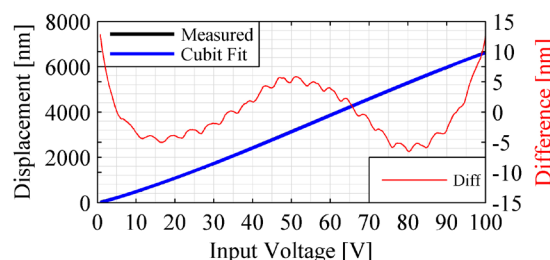


FIGURE 5. The measured displacement of both axes as a function of applied voltage from 0 V to 100 V. It is clear that the parasitic motion is approximately 50 times smaller than the piezo motion.

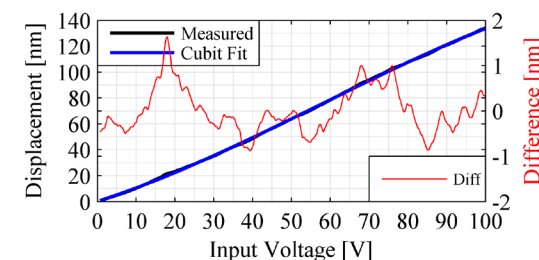
to 1 nm of motion on the Y-axis, which reduces to 20 pm of parasitic motion, a 1 nm sine wave was tested as shown in Figure 8a. Figure 8a shows that the stage is controllable to 1 nm, with a high signal to noise ratio for the Y-axis, it then follows that the parasitic motion in the X-axis should be controllable to 20 pm.

FUTURE WORK

Methods to reduce the noise must be investigated. One current limitation are refractive index fluctuations, adding noise to the measurement. In practice, to have a high bandwidth (>1 kHz), the



(a)



(b)

FIGURE 6. a.) Measured data, a cubic curve fit, and the error between the two for the Y-axis. b.) Measured data, a cubic curvefit, and the error between the two for the X-axis.

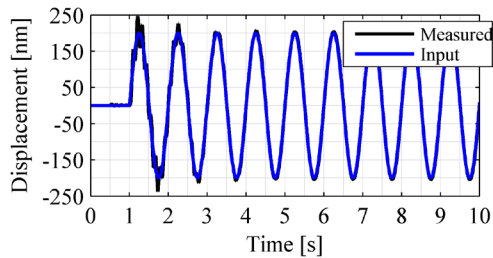
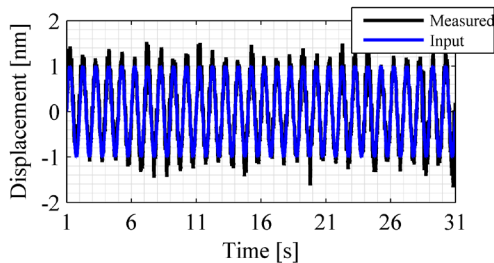
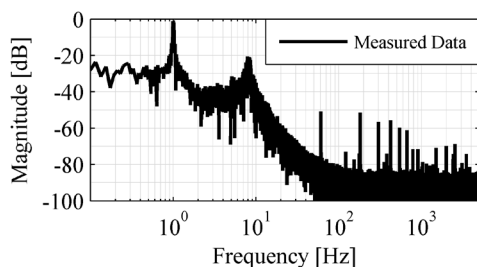


FIGURE 7. A plot of the measured displacement with a 200 nm sine wave input signal.

lock-in amplifier must have a very short time constant of approximately 0.3 ms (3.3 kHz). The refractive index noise combined with this filter setting creates problems for unwrapping the phase for measuring the stage at these bandwidths. In the laser cavity, however, the environment will be sealed and potentially in vacuum, limiting the noise from refractive index. Additionally, the feedback signal for controlling the laser cavity will be from frequency counting rather than phase unwrapping, which is less stringent than unwrapping to obtain the position. A new, FPGA-based high bandwidth data acquisition system and controller will be used as our measurement and control platform. Additionally, in order to ensure accurate control of the piezoelectric actuator when



(a)



(b)

FIGURE 8. a.) A 1 nm sine wave as measured on the Y-axis. b.) Amplitude spectrum of the data in Figure 8a

measuring only one axis, a strain gage solution is being investigated. By placing and calibrating a strain gage on the piezoelectric actuator, a controlled displacement could be applied, potentially resulting in a simplified control system. Color figures can be found in the digital copy.

ACKNOWLEDGMENTS

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Patterning Phase-Coherent Lines on Large Substrates using Direct-Write Lithography

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INTRODUCTION

Writing large blazed echelle diffraction gratings lithographically requires substantial effort be put into maintaining phase-coherence over not only the extent of the substrate but also over the long duration of exposure. The stringent requirement of less than 5 nm in low-frequency, periodic transverse (orthogonal to line direction) edge deviation while patterning 100 mm substrates [1] means that many sources of error must simultaneously be measured and, where necessary, corrected. The most obvious potential sources are the power stability of the optical source, the following error of the stage and the thermal expansion within the metrology frame. Interferometer position measurements are also subject to cosine errors and changes in refractive index due to air turbulence and low-frequency variations in air temperature, pressure and humidity. In addition, structural compliance between reference mirrors and the lithographic optical column results in both transverse error of the beam's centroid and sensitivity to defocus at the substrate resulting from our use of a high-NA objective lens. The resulting error types of greatest concern are described in Figure 1 and Figure 2 shows a cross-sectional SEM image of a test echelle grating.

To this end we have constructed a device designed to measure and correct many of the sources of uncertainty. The first version of the Line-Writer system is currently mounted on the optical frame of the Nanoruler (a Scanning Beam Interference Lithography system). MIT's recently-upgraded Nanoruler has a large 2-DOF linear horizontal stage with interferometers on each axis measuring stage position. The entire system is contained within an enclosure that has a thermal stability of ± 10 millidegree Kelvin [2]. Changes in the refractive index are tracked using a refractometer and the interferometers

are dead path corrected. Previous tests where etched gratings were rescanned in a reading mode have shown interferometer measurement precision to be within 3 nm over the course of an exposure [3, 4].

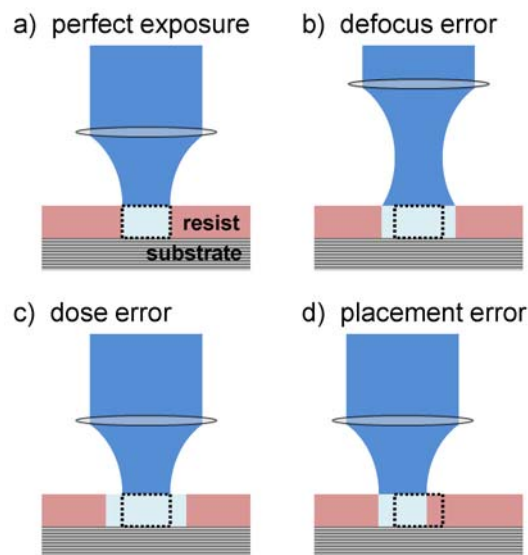


FIGURE 1. Cutaways in the transverse direction (orthogonal to the direction of motion) show the errors most likely to degrade line quality relative to the nominal exposure shown in (a). These are: defocus error (b), dose error (c) and placement error (d). The lighter areas indicate the exposed regions of resist with the dashed boxes representing the desired dose profile.

Preliminary results show promise and in the near future we expect a comprehensive characterization of the various error sources that includes an analysis of the importance of each within the context of our overall error-budget. The remainder of this document is devoted to a brief description of the recent upgrades to the Nanoruler and the design of the Line-Writer.

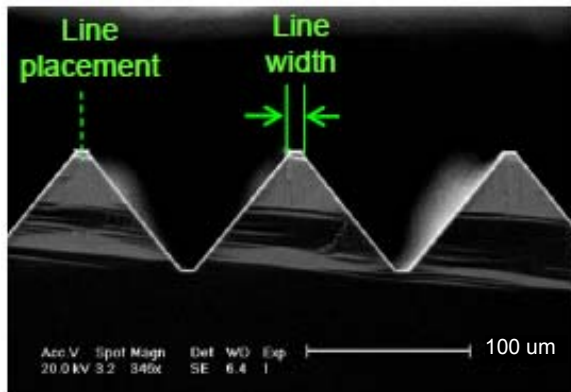


FIGURE 2. SEM image of the cross-section of a test echelle grating. Courtesy Daniel Jaffe.

NANORULER UPGRADES

Recent upgrades to the Nanoruler have focused primarily on reducing latencies and enabling the high data throughput needed for the Line-Writer and Scanning Beam Interference Lithography systems to share the same platform. We have developed an operating system that executes real-time code at a rate of 20 kHz and meets our substantial data transfer requirements. At this clock rate the RMS following error is around 40 nm, subject to the measurement uncertainty of the stage interferometers from unobservable error [3], with a dominant mode at around 100 Hz.

DESIGN OF THE LINE-WRITER

Figure 3 shows the Line-Writer system that is currently being tested. It was designed to allow both diagnosis and correction of the various sources of uncertainty that are most likely to result in low-frequency periodic errors.

The system can operate in reading mode where a diffraction limited spot ($\lambda=405$ nm) is produced on the surface of the substrate using a Mitutoyo Plan Apo NUV 50x infinity-corrected objective lens (NA=0.42). The intensity of the spot's reflection off the surface can then be measured by a mark detector (photodiode B in Figure 4) during the patterning process. Scanning across fiducial edges separating regions of differing reflectivity allows the observation of in-plane position errors between the centroid of the spot and the reference frame of the substrate.

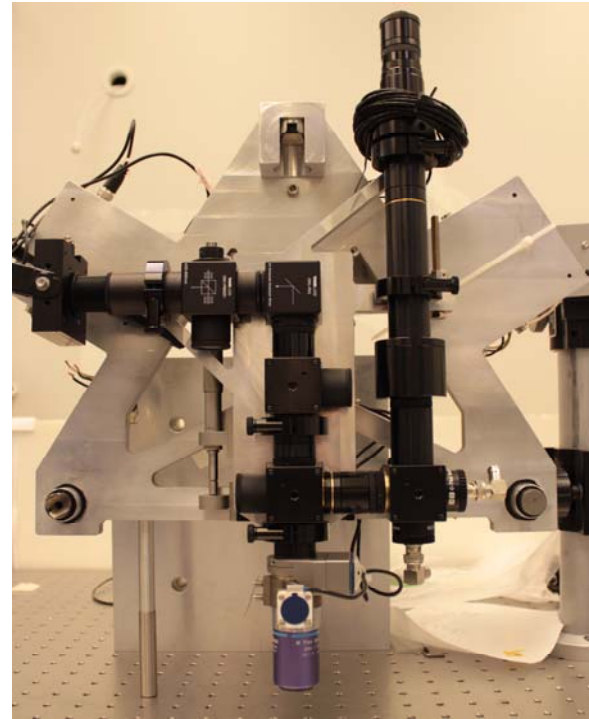


FIGURE 3. An image of the Line-Writer system.

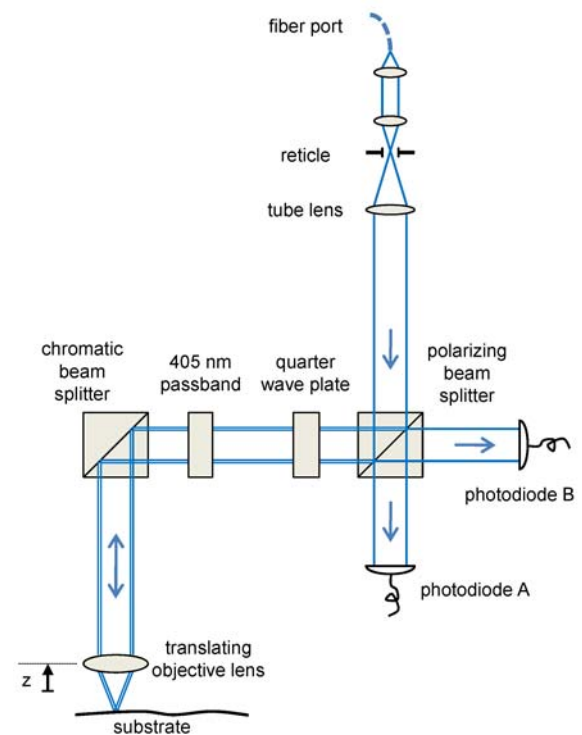


FIGURE 4. The 405 nm coherent illumination system designed to image the reticle onto the substrate with a lateral demagnification of 50x.

A beam power detector (photodiode A in Figure 4) measures the power transmitted through the reticle. The laser power is controlled by an analog signal to maintain a consistent dose throughout the exposure.

The depth of focus of the objective lens is approximately 2 microns at $\lambda=405$ nm, placing a significant requirement on depth tracking when patterning large substrates. Measurements of the distance between the objective lens and the substrate are concurrently taken by observing the defocus of the back-reflection of a 20 micron spot produced on the substrate surface. Red light ($\lambda=632$ nm) is used as to not affect the resist. The method, depicted in Figure 5, is similar to the astigmatic focus sensor used in blu-ray players. A piezo-actuated microscope objective lens mount maintains this distance to keep the image of the reticle in focus on the surface of the substrate. One of the primary benefits of this method is that the depth measurement uses the same objective lens as the light used to expose the resist. The image plane of the reticle must be kept on the same surface as that of the red light.

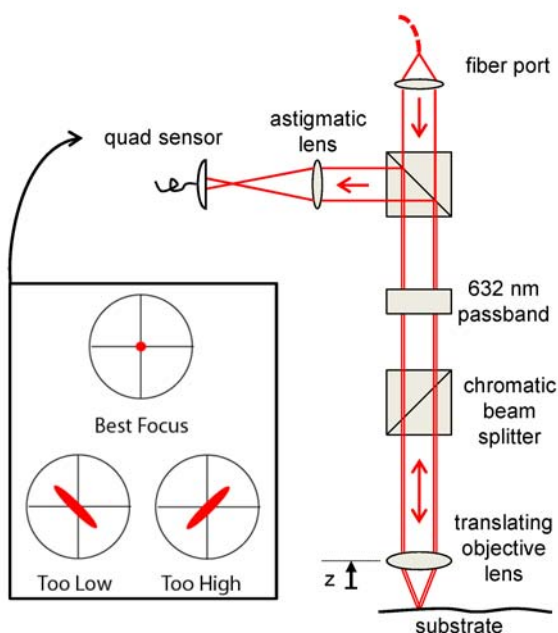


FIGURE 5. The 632 nm depth-measurement system designed to measure the distance from the substrate to the same objective lens that is used for patterning with 405 nm light.

PRELIMINARY DEPTH-TRACKING RESULTS

The results presented here show strong correlation between various measurements of depth and thus validate measurements provided by the 632 nm system. The ability to track the surface is then shown to meet the required tolerances.

A proportional controller provided acceptable tracking of a reference distance between the focal plane of the 632 nm light and the substrate surface. The stage was fixed in place by depressurizing its air-bearings. With capacitive sensors we confirmed that over short time scales, where errors due to thermal expansion could be ignored, the relative changes in the lens piezo position command agreed well with the measured displacements between the lens and the substrate. Over longer time scales drift was observed.

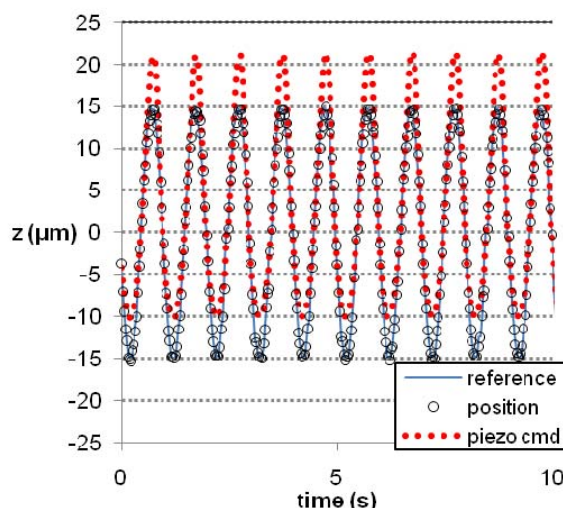


FIGURE 6. The position measured by the depth sensor closely followed a 15 micron sinusoidal reference trajectory. The commanded position of the lens piezo is also shown. The deviation of the position as measured by the depth sensor the actual position (known from the piezo position command) shows the depth sensor is only accurate when the focal plane is near the surface of the substrate.

As shown in Figures 6 and 7, when varying the reference distance quickly and observing both the closed-loop piezo position command and the output of the depth sensor, the two measurements were nearly identical within a range in depth of ± 5 microns from the surface of the substrate. With this validation of the 632 nm

depth sensor we were then able to examine our ability to track and measure changes over hours due to vibration, thermal effects, etc.

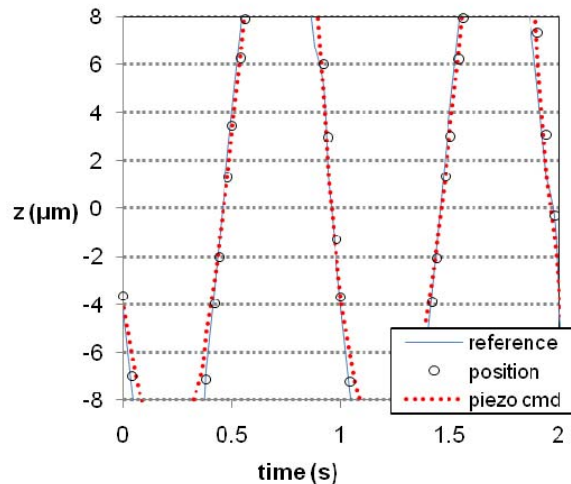


FIGURE 7. The relationship between the depth sensor position measurement and the position command of the lens piezo is approximately linear within a band of around ± 5 microns from the surface of the substrate.

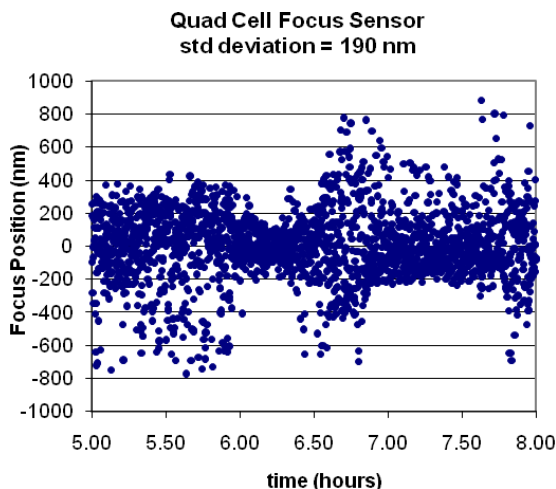


FIGURE 8. In measurements taken over 3 hours the lens was able to properly maintain its focus on the substrate surface.

As shown in Figure 8, the system was able to track the surface such that the RMS deviation of the focal plane of the 632 nm light was under 200 nm over the course of 3 hours. This is well within the depth of focus of the lens.

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Accurate Fabrication by Improvement of Lamination Path on Direct Metal Lamination Using Arc Discharge

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INTRODUCTION

Rapid prototyping (RP), which can fabricate complicated shapes from 3D-CAD data, has become popular for making prototypes. Rapid manufacturing (RM) is expected to fabricate not only prototypes but also machine parts and die tools [1].

Direct metal lamination using arc discharge has been developed by the authors' research group [2-3] as a Rapid Manufacturing (RM) technology. In this method, the metallic materials are melted and accumulated by arc welding technology. Therefore, various metallic materials can be used. The strength is comparable to the bulk material, so metallic products with high strength can be fabricated rapidly. In addition, the laminated modeling objects can be fabricated as a near net shape. So this method can reduce the quantity of the removed material and is especially expected to fabricate large-size dies and a single machine part which needs high-strength.

Generally, in RM processes, 3D-CAD data is transformed into 2D-sliced data. This data is used to make the sliced shape data for the layer manufacturing. In the direct metal lamination, the switching on and off of the welding power supply often causes the unintended steps on the manufactured surface. Therefore, general laminating processes decrease the dimension accuracy. In addition, when an inclined wall shape is fabricated, its inclined angle becomes larger than the setting angle because molten metal flows in the direction of gravitational force [5]. In this study, continuous lamination path, which does not require the switching on and off of the welding power supply, is generated from 3D-CAD data. In addition, the laminated modeling object which has inclined angle is fabricated using the continuous lamination path. Then, measured inclined angle is compared with

setting angle. Finally, the angular error is compensated by the modified lamination path.

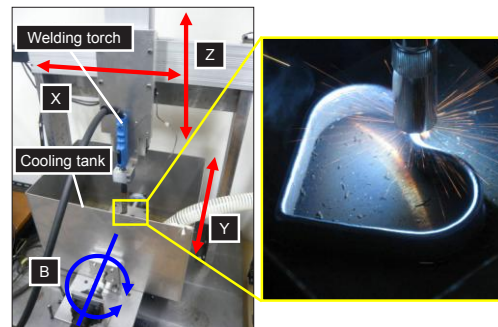
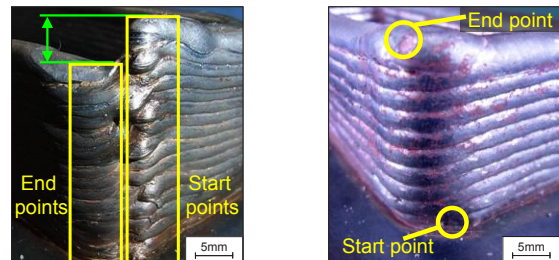


FIGURE 1. Experimental procedure



(a) Discontinuous path

(b) Continuous spiral path

FIGURE 2. Influence of the lamination path on the modeling accuracy

EXPERIMENTAL PROCEDURE

Fig. 1 shows the direct metal lamination machine. It has four axes X, Y, Z and B. A welding torch is attached on Z axis. The strokes of the X, Y, Z and B axes are 400 mm, 300 mm, 150 mm and $\pm 90^\circ$, respectively. A substrate 150 mm $\times$ 150 mm $\times$ 5 mm in size, on which the manufacturing object is accumulated, is fixed to the base block on the table. Welding wire is fed to the welding torch tip and it works as an positive electrode. The welding wire tip is melted by the arc discharge and this molten metal is dropped on the molten pool of the manufactured object. As the welding torch moves along the aimed path, accumulated molten metal is solidified and deposited. The Z-axis is controlled to keep the distance between the welding torch tip and the accumulating point to be about 5mm.

A water cooling tank is assembled as part of the direct metal lamination machine. With adjustments of the water level, the object, the base block, and the B-axis can be cooled with water.

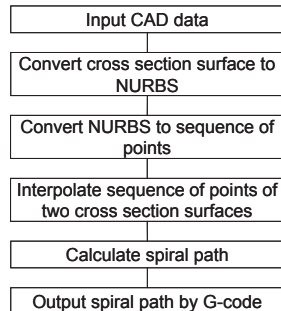


FIGURE 3. Flow chart of generating spiral path

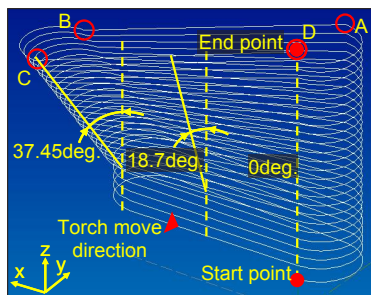
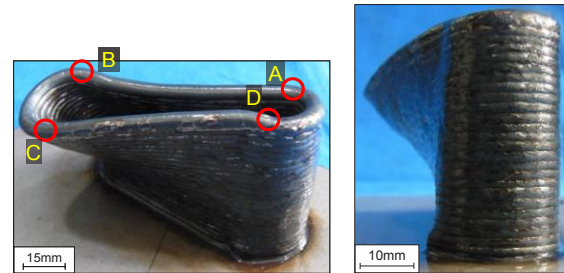


FIGURE 4. Lamination path

GENERATION OF THE SPIRAL PATH FROM 3D-CAD DATA

In the direct metal lamination, the bead height near the lamination start point is large. On the other hand, the bead height near the lamination end point is small. Therefore, the height of laminated modeling object becomes uneven with existing discontinuous lamination path as shown in Fig. 2(a). In this study, a continuous spiral path is applied to the fabrication of shell structures. The spiral path continues the lamination from the beginning of the fabrication

to the end. The spiral path can be calculated as shown in Fig. 3. With the spiral path, the height of the laminated modeling object could be fabricated without discontinuity as shown in Fig. 2(b).



(a) Overall view

(b) Right side view

FIGURE 5. Test piece

DETERMINATION OF LAMINATION PATH

Measurement of the inclined angle

In this chapter, how to improve the accuracy of the fabricated object which has the inclined wall will be discussed. By using the direct metal lamination, the laminated modeling object which has the arbitrary inclined wall can be fabricated without support material. In this case, the inclined angle is continuously varied, that is, the inclined angle gradually becomes larger or smaller as the welding torch moves. These influences were investigated. Fig. 4 and 5 show the lamination path and manufactured object respectively. In this shape, as the welding torch was moved from A to B, inclined angle becomes larger along the path. On the other hand, as the welding torch was moved from C to D, inclined angle becomes smaller. Table 1 shows experimental conditions. The influence of the welding torch feed speed and the cooling condition on the inclined angle error was also investigated. The inclined angle of the laminated modeling object was measured by a coordinate measuring machine. Fig. 6 shows the relationship between setting angle and angular

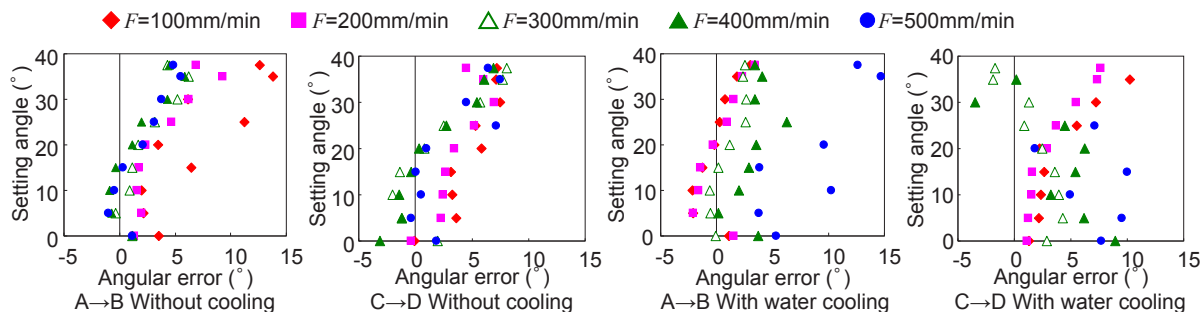


FIGURE 6. Relationship between setting angle and angular error

error. Setting angle was defined as the inclined angle from vertical line. For example, the setting angle of the vertical wall shape is 0°. Positive value of the angular error shows that the measured angle was larger than the setting angle. The maximum angular error was 15°. The error tendency depends on inclined angle, welding torch feed speed, cooling method and torch feed direction.

Table 1. Experimental conditions

Welding current	A	60
Welding voltage	V	15
Feed speed	mm/min	100, 200, 300, 400, 500
Wire material	Equivalent of SS400	
Wire diameter	mm	1.2
Shielding gas	Ar	80%
	CO2	20%
Shielding gas flow rate	L/min	5
Number of layer		20

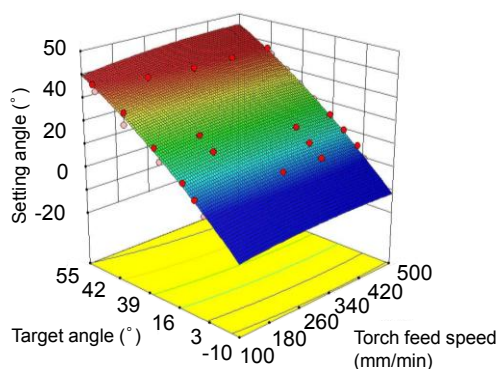
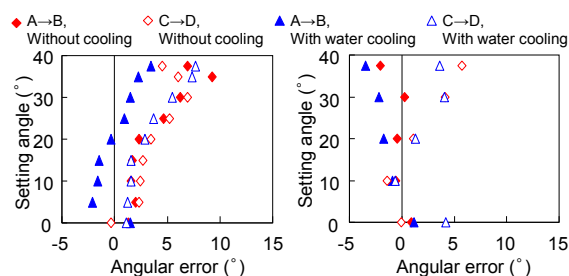


Figure 7. Response surface (A → B, Without cooling)



(a) Before compensation (b) After compensation
Figure 8. Compensation by the response surface method ($F=200\text{mm/min}$)

Compensation of the inclined angle

In order to compensate the angular error, a response surface method was applied because various factors interact and causes the angular error. Fig. 7 shows one of the response surfaces. Setting angle θ' is calculated from target angle θ and torch feed speed F with follows equations. Without cooling:

$$\theta'_{A \rightarrow B} = -5.14 + 0.980\theta + 0.0272f + 1.38 \times 10^{-4}\theta \cdot f - 3.58 \times 10^{-3}\theta^2 - 3.99 \times 10^{-5}f^2 \quad (1)$$

$$\theta'_{C \rightarrow D} = -4.22 + 0.937\theta + 0.0279f - 1.40 \times 10^{-4}\theta \cdot f - 1.62 \times 10^{-3}\theta^2 - 3.38 \times 10^{-5}f^2 \quad (2)$$

With water cooling:

$$\theta'_{A \rightarrow B} = -2.58 + 1.090\theta + 0.0345f - 2.60 \times 10^{-4}\theta \cdot f - 3.21 \times 10^{-3}\theta^2 - 7.94 \times 10^{-5}f^2 \quad (3)$$

$$\theta'_{C \rightarrow D} = -2.68 + 0.917\theta + 0.0169f + 1.23 \times 10^{-3}\theta \cdot f - 5.51 \times 10^{-3}\theta^2 - 8.02 \times 10^{-5}f^2 \quad (4)$$

Then, the laminated modeling object was fabricated by the modified spiral path. Torch feed speed F was 200mm/min. Fig. 8 shows the result of the compensation. From this figure, 22% of the angular error was decreased.

Discussions

As shown Fig.8, the angular error, when the setting angle was equal to or lower than 20°, became small after compensation. However, the effect of the compensation for the angular error when the setting angle was larger than 20° was small. Here we discuss the reason of these results. As shown Fig. 9, the lamination height of the area, where the setting angle was large, was decreased. In the area where setting angle was 0°, arc position was at the center of the bead where the lamination height is the largest. Therefore, the lamination height of the molten pool and the next layer become high. On the other hand, in the area where setting angle is 37.5°, arc position shifts from the center of the bead. Thereby, the lamination height of the molten pool and the next layer becomes low. For this reason, the lamination height error increased layer by layer and finally, this error became 8mm. When the lamination height becomes lower than expected, the distance between the welding torch and laminated modeling object becomes large. Thereby, as shown in Fig. 10, the length of the welding wire from the welding torch tip to the arc point became long. it caused unexpected error of the arc position and the laminate position because the welding wire was curved. In the area where the setting angle is large, the lamination height error gradually increased and the welding wire length became large. The arc position and the laminate position error gradually increased. They caused unexpected angular error. For this reason, the lamination height should be controlled according to the setting angle. The lamination height can be controlled by changing

the welding current and torch feed speed. Especially, torch feed speed can be controlled easily by NC program.

The angular error tendencies that observed on the path from A to B and from C to D were different. Molten metal forms the bead with being influenced by gravity, molten metal flow and surface tension. By the difference of the welding torch feed direction, the molten metal flow was changed. To minimize the difference of the angular error tendency caused by welding torch feed direction, lamination height should be controlled, and the molten pool should be kept horizontal.



Figure 9. Test piece after compensation (Front view)

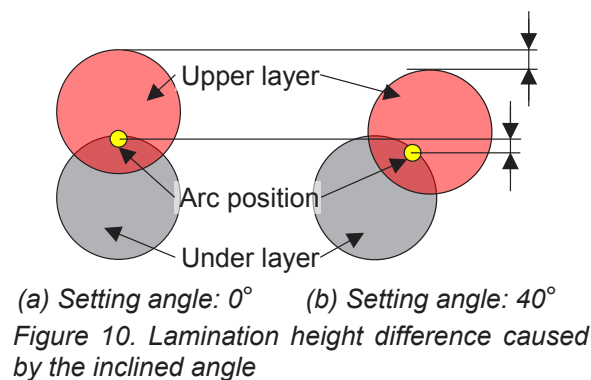


Figure 10. Lamination height difference caused by the inclined angle

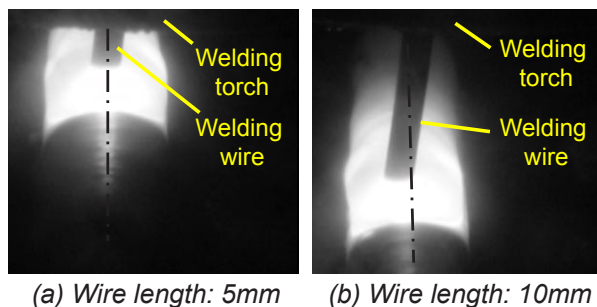


Figure 11. Arc position difference caused by wire length

CONCLUSIONS

In this paper, the shell structures which have inclined angle were fabricated by direct metal lamination. And its inclined angle was measured and compensated. Following conclusions were obtained.

- 1) With the continuous spiral path, the laminated modeling object can be fabricated with high accuracy.
- 2) The angular error tendency depends on inclined angle, welding torch feed speed, cooling method and torch feed direction.
- 3) The angular error was decreased 22% by the compensation using the relationship obtained by the response surface method.
- 4) The effect of the compensation was small when setting angle is larger than 20° . The angular error tendencies that observed on path from A to B and from C to D are different. To fabricate with higher accuracy, the lamination height should be controlled by changing torch feed speed according to the setting angle.

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MACHINING AND GRINDING CRITICAL COMPONENTS FOR THE MARS ROVER, *CURIOSITY*

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INTRODUCTION

On November 26, 2011 NASA launched the Mars Science Laboratory's rover, "Curiosity." A part of Curiosity's features is an arm with a 7.5-foot reach that is driven by actuators built from 14 components machined by Andrew Tool & Machining and ground by Professional Instruments Company. The \$2.3 billion project's objective is to evaluate the planet's past and present ability to support life. Curiosity is larger, can travel further, and carries ten scientific instruments, more than its predecessors, Spirit and Opportunity.

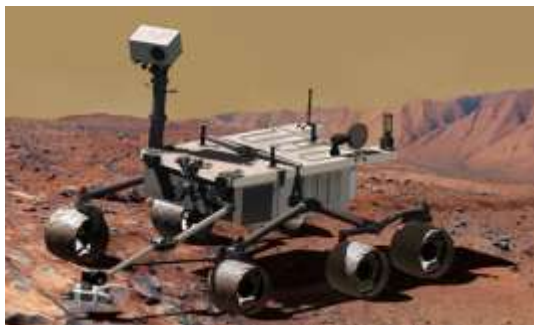


FIGURE 1. Artist's conception of the Mars Rover, *Curiosity*.

ANDREW TOOL & MACHINING

Andrew Tool & Machining specializes in effectively machining precise and complex parts. The Minnesota-based company has a reputation for performing extremely tight tolerance machining on production parts.



FIGURE 2. Andrew Tool & Machining, Plymouth, MN.



FIGURE 3. At Andrew Tool, a Hermle 5-axis VMC machines the complex actuator housing.

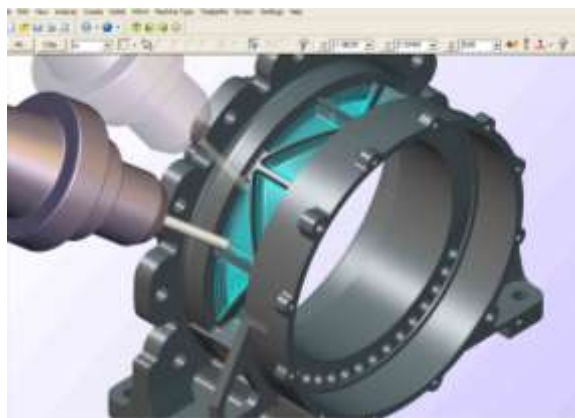


FIGURE 4. Andrew Tool used Master CamX4 to generate the machine tool paths.

PROFESSIONAL INSTRUMENTS

Professional Instruments Company is known primarily as an air-bearing spindle manufacturer, but we also do a fair amount of precision subcontract work. Professional Instruments was asked by the Jet Propulsion Laboratory to assist in finishing the critical mobility system components for Curiosity. We got involved because of the thin-walled nature of the parts and the tight geometry tolerances.

We become involved in projects like this when conventional grinding techniques don't produce the results required. With over 60 years of experience to draw on, we are uniquely qualified to work on projects like this. The design required sub-micron roundness and dimensional tolerances; the bearing pockets had to be concentric to the gear teeth. The parts are made out of Vascomax tool steel and were designed with such thin walls that standard part holding tools like scroll chucks were not up to the task.

In spite of the many challenges involved in this project, we were able to grind the parts to the required specifications and learned many important lessons in the process.

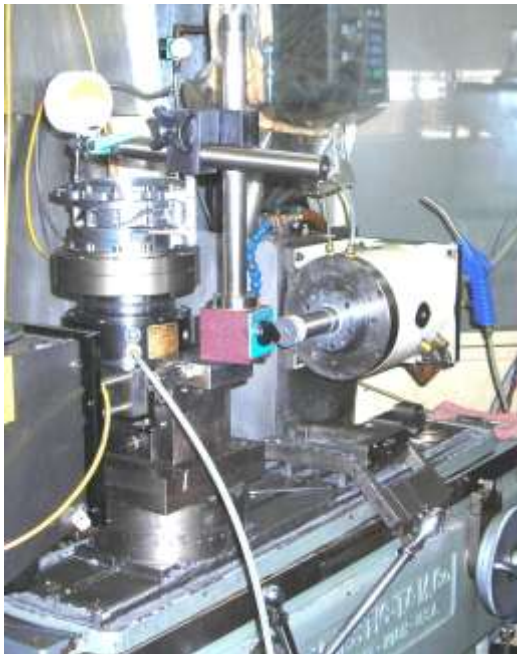


FIGURE 5. The thin-walled nature of the parts required us to load them in a vertical orientation and then tip them down for grinding.



FIGURE 6. Spindle tipped down for grinding.



FIGURE 7. We ground all mounting surfaces and bearing pockets to precisely align them to the pitch diameters on the gears.



FIGURE 8. Roto grinding allows us to hold flatnesses 0.5 micron or better on surfaces.



FIGURE 9. Completed actuator unit.

Tim Sheridan lost his battle with cancer in May of this year. This project would not have been possible without his keen insights into making the bearing pockets concentric to the gear pitch. He designed the innovative tooling and personally oversaw much of the fine grinding of these difficult to manufacture parts.



Tim Sheridan

METROLOGICAL SOLUTIONS FOR DETECTIVE OPTICS MANUFACTURE

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Nowadays, illumination systems need to achieve some characteristics regarding performance, energy efficiency and production cost.

Stamped refractive structures on thermoplastic substrates, spread and distribute the light, and therefore an optimization of the luminous efficacy is done.

The FlexPAET project aim is the development of an innovative process chain for the high volume production of large area micro structured surfaces in thermoplastic substrates.

The idea is to manufacture a master substrate using adaptive embossing technology to stamp the micro structures. This process need to be tested to ensure that the final results are the desired ones, and here is where metrology takes part.

Using an in-line metrology system it is possible to test dimensional parameters (structures and stamping tools) and functional parameters (luminosity and directivity of the emitted light), to decide if the master substrate needs to be reworked or if it is ready for the replication operations.

A custom-build metrology system has been developed successfully for the FlexPAET samples inspection and has been integrated to the embossing machine and automated in order to achieve the production necessities.

The result is a real time inspection system of the embossing process that provides feedback to the control system and also provides information about all the requested measurements by the control, in order to avoid stamping errors and to prevent future reworks.

TEMPERATURE MEASUREMENT AND STUDY OF THE THERMAL EFFECTS DURING THE MICRO-LASER ASSISTED MACHINING PROCESS

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INTRODUCTION

This paper presents a technique to measure the temperature (and thereby energy absorption) due to laser irradiation of Silicon (Si) and its effect on the pressure-induced metallic phase formation during the Micro-Laser Assisted Machining (μ -LAM) process. μ -LAM is a process used to achieve atomic-level surface smoothness of hard-to-machine semiconducting materials and ceramics while increasing the material removal rate (MRR) and reducing tool wear. This is achieved, as shown in Fig. 1, by machining the surface of these materials, in a ductile regime, with an optically transparent diamond tool while a laser beam is passed through the tool. Through the formation of the high pressure phases beneath the tool due to preferential heating, μ -LAM has been demonstrated to enhance the ductility and reduce the brittleness of these materials; thus, enabling a higher MRR, good surface finish, and higher life of the cutting tool [1].

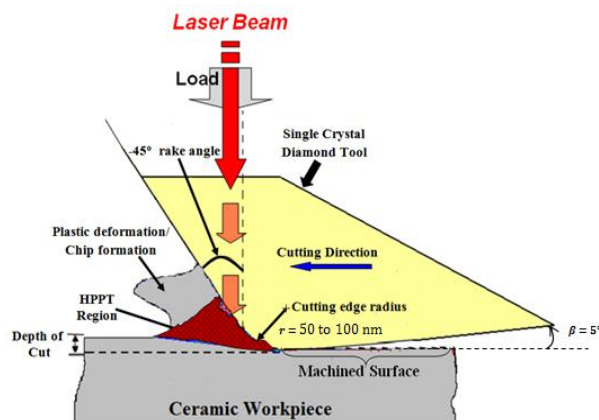


FIGURE 1. Schematic cross-section of the μ -LAM system [1]

During μ -LAM, the MRR is increased by optimizing the machining parameters, i.e., normal force applied/depth of cut, cutting speed, laser power, feed, and wavelength of the laser. Fig. 2 shows the physical set up of the μ -LAM system on a Universal Micro-Tribometer from CETR Corporation. In this setup, a laser is guided through a fiber and coupled to a Beam Delivery Optics unit which converges the beam to 10 μ m in diameter and focuses it on the cutting edge of the tool. A 100 W, CW, 1070 nm- wavelength fiber laser is used in this setup.

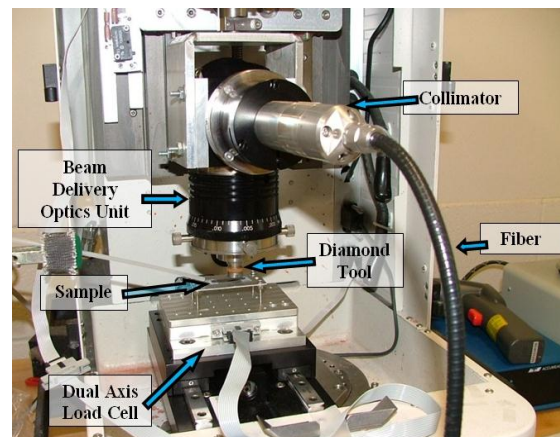


FIGURE 2. Setup of the μ -LAM system on Universal Micro-Tribometer [1]

MOTIVATION FOR THE TEMPERATURE MEASUREMENT DURING μ -LAM

In order to optimize the μ -LAM process, a temperature measurement of the machined region where high pressure phase transformations (HPPT) occur, is desired. This temperature measurement provides an idea of the amount of energy absorbed in this region. Previous work aimed at measuring/predicting

this temperature includes the use of a thermal camera [2] and the use of analytical and numerical models solved with the help of heat conduction equation for a semi-infinite solid over time with a Gaussian laser beam intensity distribution [3]. Despite the fact that these two methods did not directly measure the temperature of the HPPT region, they did yield useful information. The use of the numerical and analytical models helped estimate the temperature of the HPPT region of Si (500 °C - 800 °C), and the use of the thermal camera demonstrated that the HPPT region is the area that absorbs the majority of laser radiation during the μ -LAM process. An *in situ* temperature measurement of this region possesses three main challenges that make it difficult to perform. These challenges are: the size of the HPPT region (around 50 μ m X 50 μ m), the fact that this is an unexposed region, and finally temperature has been estimated to be high (greater than 500 °C). This paper attempted a new *in situ* temperature measurement technique that correlates the sensed temperature in the back of a wafer to the actual temperature of the HPPT region.

FABRICATION PROCESS OF RESISTANCE TEMPERATURE DETECTOR (RTD) AND EXPERIMENTAL SETUP

The temperature measurement was done on a Si wafer by reducing its thickness and subsequently depositing a thin film Platinum RTD on its backside. The thickness of the wafer is reduced to minimize the conduction effect through the wafer.

The thickness of the Si wafer is reduced from 550 μ m to 104 μ m through lithography and the final required thickness was determined using FEA analysis based on the intended applied load. The Platinum RTD is deposited at the back of the thinned wafer to measure the HPPT zone below the cut surface. Finally, the thermal heat flux equation for a thin plate is used to account for the conduction through the thickness of the wafer. Fig. 3 shows the cross-sectional view of this experimental approach.

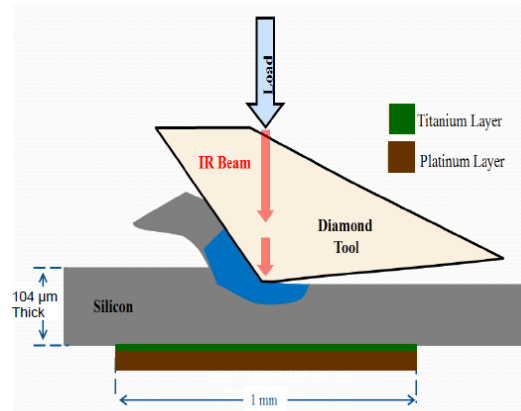


FIGURE 3. Cross-section of the μ -LAM Process with thin film RTD

Lithography & Wet Etching

To achieve a thickness of 104 μ m, the Silicon wafer is exposed to a conventional lithography process, followed by etching in a 30% KOH bath at 90 °C for 100 minutes. The roughness (R_a) of the final etched surface was measured to be less than 30 nm. Fig. 4 shows the trench pattern on the wafer after lithography; the RTD is constructed on the opposite side of the wafer.

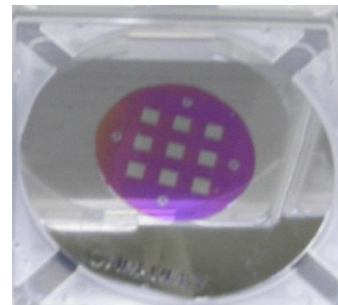


FIGURE 4: Micromachined Silicon wafer after lithography (104 μ m Thick square trenches)

Physical Vapor Deposition (PVD) and Wire Bonding

A pattern consisting of six identical rectangles (1 mm X 5 mm) is deposited at the back of the wafer through PVD using a shadow mask. A 25 nm Titanium (Ti) layer is deposited to assist in the adhesion of the Platinum layer. Next, a 50 nm layer of Platinum is deposited on the Ti layer to serve as the sensor material given its high temperature-to-resistance sensitivity. Finally, a 500 nm layer of Gold is deposited on the contact pads to allow for the leads to be bonded to the sensor. Gold micro-wires ($\approx$ 18 μ m diameter) are bonded to the Gold and Copper pads and finally gage wires are soldered to the Copper pads to connect to the data acquisition system. Fig. 5

shows the RTD pattern with the gage wires bonded to the Copper pads, which connects to the data acquisition system.

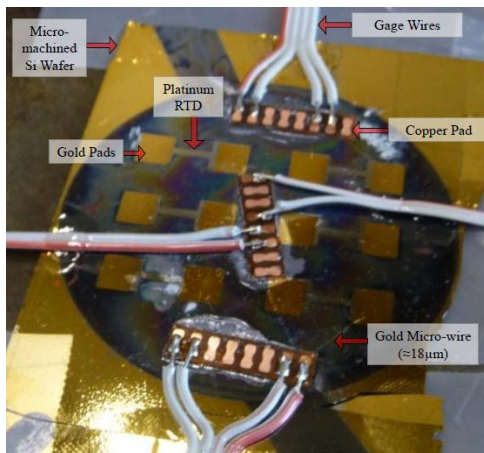


FIGURE 5. The 104 μm -thick wafer and RTD with leads bonded

Data Acquisition

Fig. 6 shows the instrumentation setup for the temperature measurement using the RTD. This instrumentation setup includes a voltage divider circuit, a computer with a data acquisition card and LabVIEW program installed, and an analog breakout accessory (the BNC 2090) to connect the leads from the RTD to the computer.

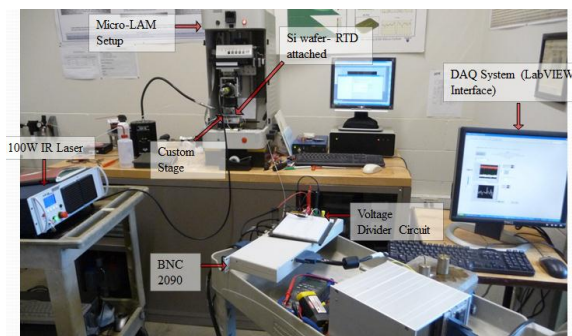


FIGURE 6: Instrumentation setup for temperature measurement with RTD

Care was taken while measuring temperature to reduce the sources of error. For example, the excitation voltage was set relatively low ($\sim 10\text{ V}$), i.e., to avoid increasing the temperature of the RTD itself which would have introduced error due to self-heating.

Fig. 7 shows the front panel of the LabVIEW program that was written to measure the resistance from the RTD. When the VI is run, it plots the resistance of the RTD on a chart (titled

RTD Resistance Monitoring). The resistance measured is converted to the corresponding temperature after the RTD was calibrated (assuming a linear response from the RTD).

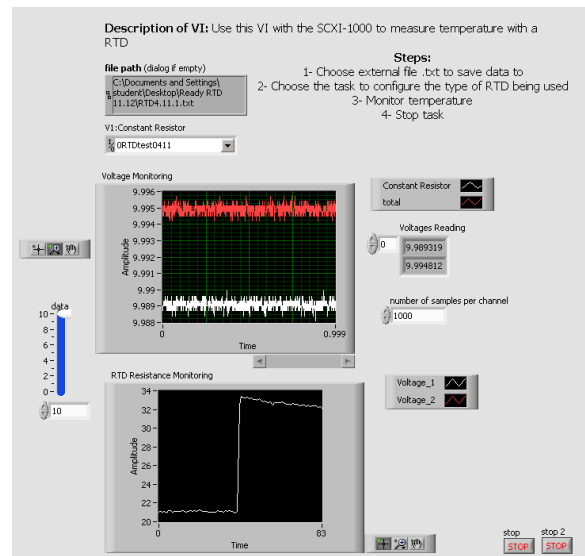


FIGURE 7: Front panel of VI constructed to measure resistance from the RTD

EXPERIMENTAL PROCEDURE

A machine load of 100mN was used for the cut performed across the opposite side of the sensor. The cut was 2 mm long with a cutting speed of 5 $\mu\text{m/s}$. The test was initiated with the laser power set to 0%; this technique was used in order to compare the effect of the laser heating with and without applied laser. Laser power was varied during the machining process, to illustrate the effect of increase in laser power on the temperature of the HPPT region.

The data recorded during the RTD experiment can be seen in Fig. 8. With no laser power and the cut in process a temperature of approximately 24°C was recorded as expected at room temperature. At a time of 50s the laser was powered to 10W. This laser heating resulted in increasing temperature of 225°C. Once reaching this temperature, there was a steady and small decrease in temperature as the cut continued. This relationship shows that there is no sensor saturation effect. The steady drop in temperature can be correlated to higher amounts of energy being expended to continuously create the HPPT region as the tool moved across the sensor.

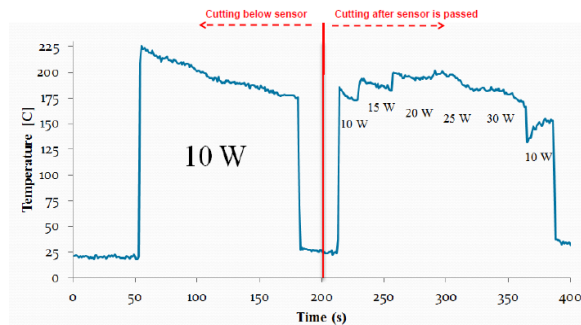


FIGURE 8: Temperature measured by RTD at different laser power

To correlate the temperature recorded by the RTD on the opposite side of the HPPT region, a thermal conduction calculation was performed to determine the actual temperature of the HPPT region. Using Equation 1, the temperature of the HPPT region was calculated to be 780°C. A 2W effective power at the tip of the tool was used for this calculation and the area of the HPPT was approximated to be 50 μm X 50μm.

$$T = \frac{\left(\frac{P}{A} * L\right)}{k} + T_m \quad (1)$$

CONCLUSION

A technique, sensitive enough to correlate the temperature of the HPPT region of Si, was successfully developed. The response of the RTD sensor was ideal and illustrated the relationship between the power of the laser and the temperature of the HPPT region. Using a thermal conduction equation, the measurement made from the backside of the wafer was correlated to the temperature of the HPPT region of Si and found to be around 780 °C at 10 W of laser power.

FUTURE WORK

Future improvements of this design could implement an array of thin film thermocouples, with hot junctions sizes comparable to that of the HPPT region; thus, enabling to obtain a temperature gradient along the profile of the cut. The research on the μ-LAM process can be extended to study the energy absorption of the metallic phases during the machining of different semiconductor and ceramic (brittle) materials. .

ACKNOWLEDGEMENTS

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Facility at University of Michigan is acknowledged for lithography and RTD patterning work.

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SUPERFINISHING PERFORMANCE OF BEARING STEEL WITH CeO₂ SUPERABRASIVE STONE

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ABSTRACT

In this paper, we investigated the superfinishing performance of bearing steel with vitrified bonded mechanochemical stone composed of CeO₂ abrasives, which can act as oxidizing agents. Superfinishing experiments showed that finishing mechanism of CeO₂ stones changed when superfinishing pressures exceeded boundary. On the basis of this finding, superfinishing performance of CeO₂ stones of three grades, i.e. I, T, and X, were evaluated through the superfinishing of bearing steels. Experiments showed that CeO₂ stone of grade I is superior in surface finish to other two grades because of the mechanical action of the crushed bonds.

INTRODUCTION

Mechanochemical superabrasive stone containing CeO₂ produces less subsurface damage and smaller compressive residual stress, smoother surface with negative skewness than conventional stones because of synergistic effect of mechanical action of hard abrasive and chemical action of the CeO₂ abrasive [1]. The stone composed of CeO₂ abrasive (CeO₂ stone) has different structures to conventional stones (FIGURE 1). It seems that agglomerate soft CeO₂ abrasive are dispersed into the bond posts. When superfinished with lower stone pressure, the CeO₂ stone truncates the roughness of pre-finished surface by rubbing. On the other hand, when superfinished with higher stone pressure, finishing rate and surface roughness improves because of the mechanical action of the crushed bonds.

In this paper, we investigated the superfinishing performance of bearing steel with vitrified bonded CeO₂ stones. First, the critical pressure of each stone was examined by considering the relative speed and the grade of stones. Second,

basic experiments were performed to check whether the higher pressure could produce better surface. And last, in order to examine which stone could perform the best, superfinishing experiments with higher stone pressure were conducted using these three stones.

SUPERFINISHING MECHANISM OF CeO₂ STONE

FIGURE 2 shows SEM micrographs of the CeO₂ stones. In grade I stone, it seems that small bond posts are anchored each other, while grade T and grade X stone, it seems that larger

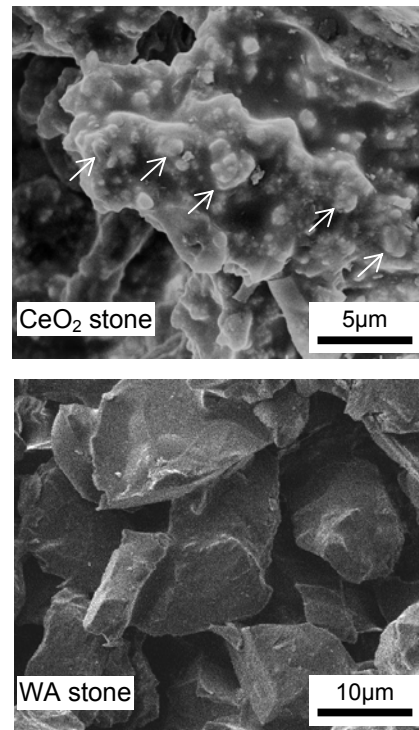


FIGURE 1. Structure of CeO₂ stone and WA stone.

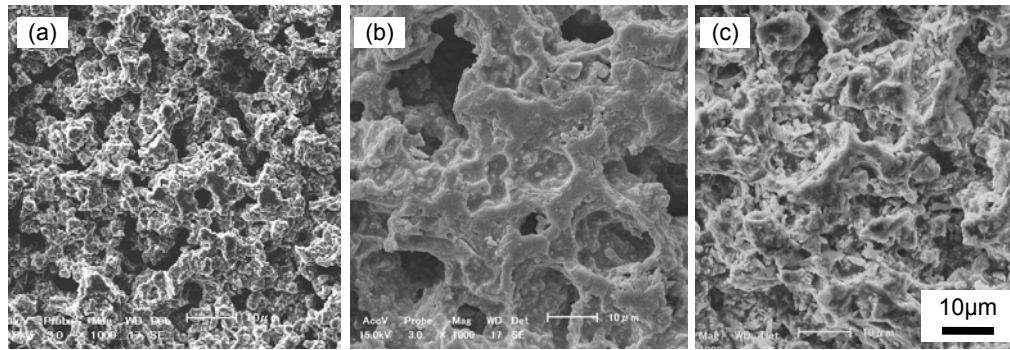


FIGURE 2. SEM micrographs of CeO_2 stones after dressing.
(a) Grade I stone, (b) grade T stone, and (c) grade X stone.

and smoother bonds containing CeO_2 abrasives are bridged.

FIGURE 3 shows the comparison of boundaries of superfinishing mechanism of grade T stone. Each relative velocity between stone and workpiece has its boundary of changing the superfinishing mechanism from mirror finish to cutting finish, which made scratches on finished surface.

FIGURE 4 shows the boundary of superfinishing mechanism of grade T stone. Black circles indicate stone wear and scratch generation, and while circles indicate mirror finishing. Each stone has each boundary of superfinishing mechanism.

SUPERFINISHING PERFORMANCE OF CeO_2 STONES

All tests were performed on a flat superfinishing machine. The end of the workpiece was secured to the work spindle, then face-finished by the rotating cup-shaped abrasive stone. During the superfinishing process, water-based superfinishing fluid, consisting of an emulsion of 1 % rust inhibitor, was used as a coolant that would exclude any chemical reaction. TABLE 1 tabulates the superfinishing conditions that were used. FIGURE 6 shows the comparison of pre-finished surface and finished surface with grade T stone for 180 s. The roughness profile is truncated and has clearly asymmetrical height distribution. The feature of the superfinished surfaces generated by the stones of different kinds is also reflected in the roughness, as seen in FIGURE 7. FIGURE 5 shows SEM micrographs of the CeO_2 stones after superfinishing time 180 s. Grade T and grade X stones were worn on their working area. In contrast, the working area of grade I stone changed because of repeated crush of bonds.

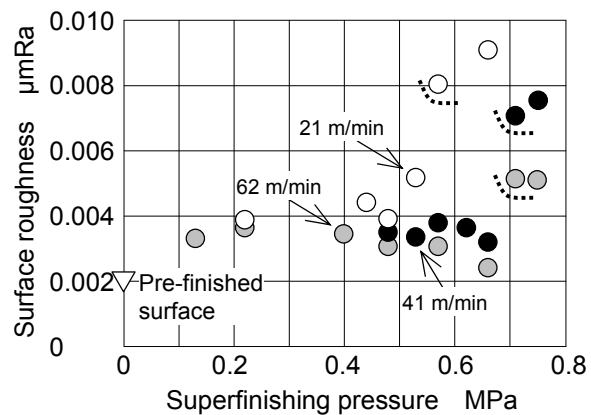


FIGURE 3. Comparison of boundaries of superfinishing mechanism (grade T stone).

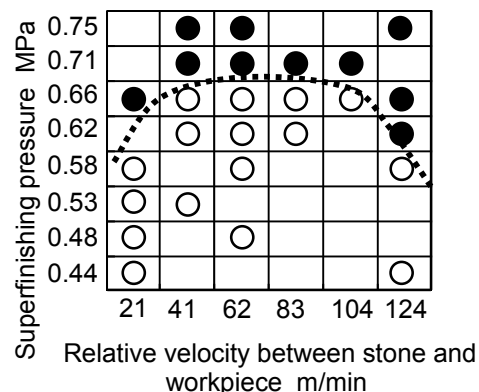


FIGURE 4. Boundary of superfinishing mechanism of CeO_2 stone (grade T stone).

FIGURE 8 shows the comparison of working area ratio in superfinishing. As the superfinishing time getting longer, the working area of grade T stone and grade X stone became larger. Meanwhile, the working area of the grade I stone fluctuated while superfinishing because of repeated crush of bonds.

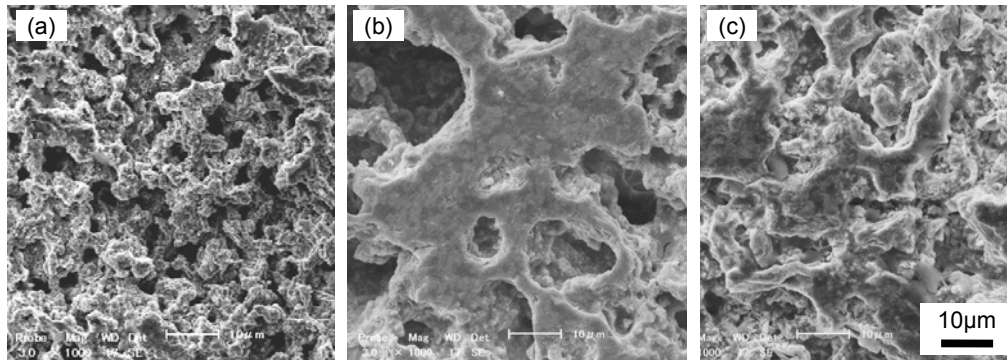


FIGURE 5. SEM micrographs of CeO_2 stones (superfinishing time 180 sec).
(a) Grade I stone, (b) grade T stone, (c) grade X stone.

TABLE 1. Superfinishing conditions.

Superfinishing pressure	0.66 MPa
Stone speed	15.7 m/min
Workpiece speed	165 m/min
Oscillation	16.7 Hz
Amplitude	0.5 mm
Superfinishing time	1 – 180 s
Superfinishing fluid	Water (rust inhibitor 1%)
Workpiece	SUJ2, HRC 60

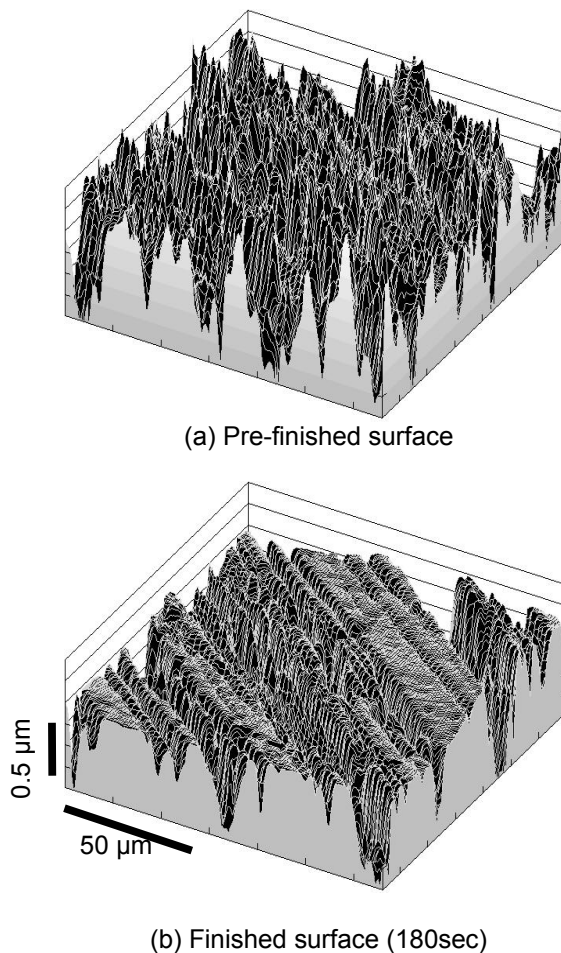


FIGURE 6. Comparison of pre-finished surface and finished surface.

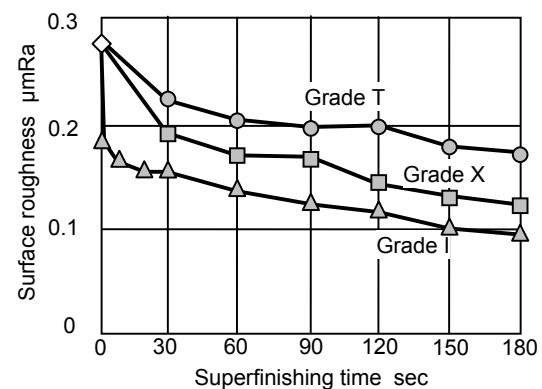


FIGURE 7. Roughness change in superfinishing of bearing steel with each stone

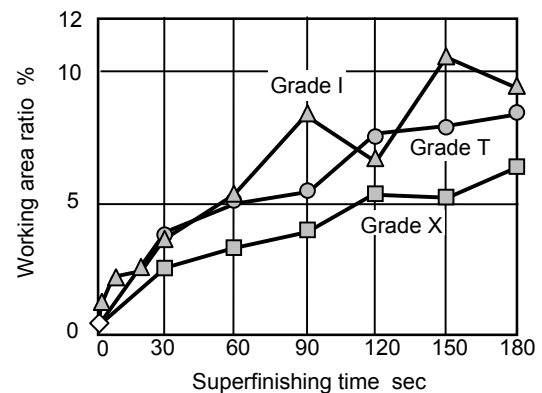


FIGURE 8. Working area ratio change in superfinishing of bearing steel with each stone.

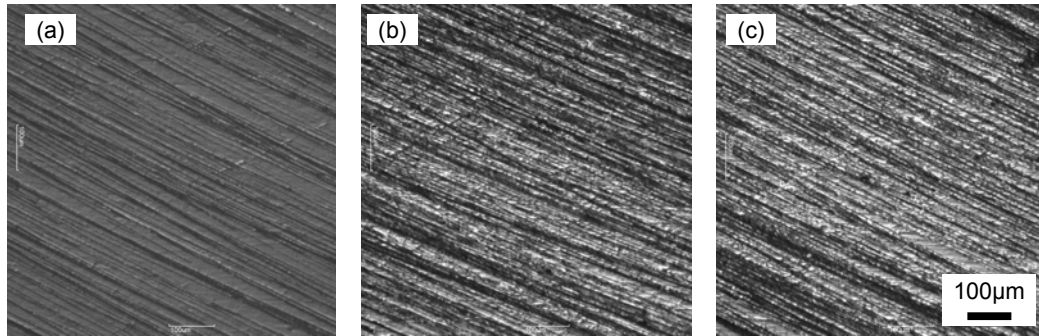


FIGURE 9. Finished surface of bearing steel (superfinishing time 180sec) with (a) Grade I stone, (b) grade T stone, (c) grade X stone.

FIGURE 9 shows superfinished surfaces of bearing steel with each stone. As shown in FIGURE 6, the roughness profile is truncated and has an asymmetrical height distribution. The skewness, which is a third-order measure of the symmetry of a profile, offers the feature of the profile. FIGURE 10 shows the comparison of skewness of superfinished surface. It is found that the grade I stone generates the most negative skewness, which means the stone generates smooth surface with valleys.

FIGURE 11 shows the schematic drawing of finishing mechanism of CeO_2 stones with lower superfinishing pressure and higher pressure. When superfinished with lower pressure, soft CeO_2 abrasives oxidize the surface and the stone truncates the roughness by rubbing. In superfinishing with higher pressure, on the other hand, the crushed bonds truncate the roughness of pre-finished surface effectively because of its mechanical action, cutting ability.

CONCLUSIONS

- (1) CeO_2 stones produce smooth surface that is truncated, but has valleys.
- (2) In superfinishing with higher pressure than critical pressure, the bonds crushes and truncate the roughness effectively.

ACKNOWLEDGEMENTS

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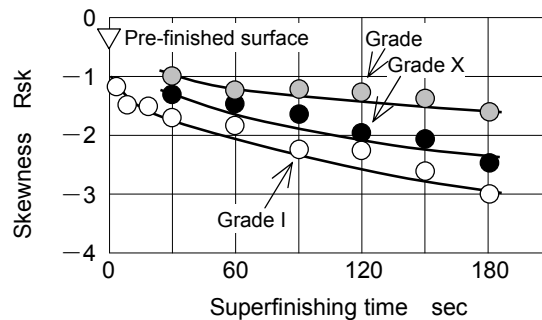


FIGURE 10. Comparison of skewness of steel workpiece superfinished with three different stones.

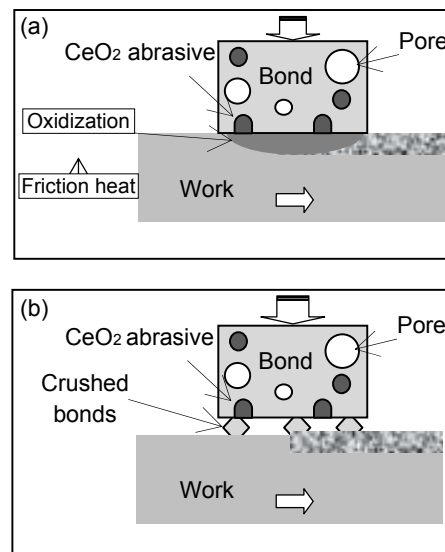


FIGURE 11. Finishing mechanism model of CeO_2 stone based on (a) oxidization and (b) crushed bonds.

METHOD FOR MEASURING THERMAL DISTORTION OF LARGE MACHINE TOOLS

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SUMMARY

Today, as the demand for large parts increases, accordingly large machine tools have got greater importance; a clear example is the production of wind turbines, which are growing in size in each new model. Thermal distortion is one of the principal factors among those that limit positioning accuracy in large machines. Therefore measuring thermal distortions for subsequent compensation is one of the greatest challenges for the machine tool industry today.

To further minimize thermal effects, it is necessary to know first which the thermal behavior of the machine is. Large scale metrology equipment such as laser tracking devices provides great flexibility and long range which are useful for machine set up, but the uncertainty given by this kind of equipment is not low enough to perform thermal distortion measurements with the required accuracy. However, laser interferometer multilateration schemes improve greatly measurement accuracy.

For that reason a new method for the characterization of thermal distortions of large machine tools is presented which solves the difficulties associated with this kind of measurements. Main benefit of this method is being able to measure thermal distortion of the toolpoint in all the working volume of large machine tools up to several cubic meters.

In addition precision electronic levels are used to directly measure the inclination of different parts of the machine and also temperatures are recorded in critical points related to the main heat sources of the machine.

The automation of the measurements provides the adequate cadence of data acquisition to check the evolution of the thermal distortion over time as the machine heats up under different working conditions.

The aim of this paper is to present the capability of proposed method to obtain measurements that could be used to perform a thermal characterization of the machine tool that will not be covered within this work.

INTRODUCTION

In the past, several methods have been used to assess the thermal distortion of large machine tools. Thermal distortion measurement is regulated under international standards for conventional measurement techniques such as laser interferometer for thermal distortion caused by moving linear axes, capacitive, inductive or retractable contacting displacement sensors for environment testing and thermal distortion caused by rotating spindles [1]. The use of invariable metrological reference artifacts and inductive displacement sensors allows measurement of the thermal distortion at the tool tip, discriminating the thermal distortion from geometrical and kinematic errors of the machine. This kind of artifacts has been used before, for calibration of CMMs, either in 2D configuration like a grid plate [2], or in 3D configuration like a spatial frame of tetrahedral form [3]. This method is also worth of use in workshop environment because of the low cost, easy set up and robustness. This makes it valuable for performing periodical characterization by machine user. However only small volume dimensions can be reached and bulky metrological reference artifacts have to be handled.

More recently, large scale metrology equipment has been developed such as laser tracking interferometer devices. These tools provide great flexibility and long range which are useful for machine set up, but the accuracy given by this kind of equipment is not low enough to perform thermal distortion measurements with the required resolution. Laser multilateration schemes performed with laser tracking devices greatly improve measurement resolution [4]. To the present this strategy has only been conducted to make static volumetric accuracy measurements in large machine tools [5]. However, the necessity of performing several sequential measurements from different locations makes not possible to record thermal distortion with the required sampling rate. This could be solved by the simultaneous use of

several laser tracking equipment units, but this is not realizable in a practical manner because of its high cost.

PROPOSED METHODOLOGY

Measurement of the volumetric positioning deviation of the tooltip

For this purpose a single commercial laser tracking interferometer was used. Multilateration method was applied being the inputs the theoretical machine coordinates and the interferometer measurements including its associated uncertainty.

The mathematical equations are linearized and solved to obtain measured machine toolpoint coordinates and laser stations positions with its associated uncertainty. For performing the multilateration, the laser tracking unit has to be located in different positions that will be called henceforth laser stations. A set of different laser stations gives a single multilateration measurement that provides the position of the tooltip [6].

For measuring different machine thermal states over time, this multilateration measurement has to be repeated periodically. To achieve a sufficient set of multilateration measurements over time two requirements are necessary:

- i) repetitive positioning of the used laser stations,
- ii) minimal time spent in the translation from one laser station to the next one.

The way in which this problem has been solved constitutes one of the main innovative features of the proposed methodology. The laser tracking unit is put on the machine table, which has the capability to translate and rotate controlled by the CNC. This configuration can be seen in figure 1.

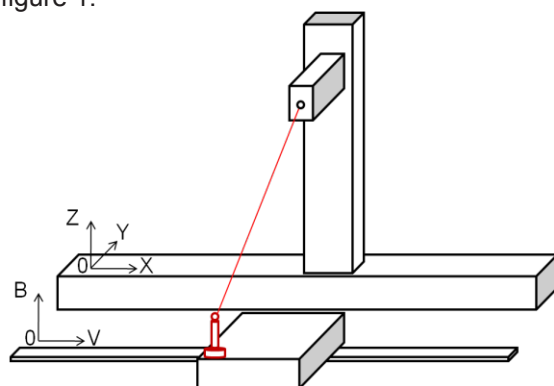


FIGURE 1. Machine under study, showing coordinate system used.

Uncertainty given by these multilateration measurements were proven sufficient to study

the thermal behavior of the machine. The continuity of the measured points and the smooth tendencies that can be seen in the results would make possible to assess the characterization of the thermal distortions.

Measurement of the angular deviation

For measuring the angular deviation a set of four digital precision wireless levels was used, two in both axes of the RAM and the other two in both axes of the column. This provides a direct measurement of the magnitude to be monitored, with 1 μ m/m resolution.

Measurement of temperatures

For doing the temperature measurements Resistance Temperature Detectors (RTD-s) are used which provide the required resolution (0,1°C). The location of the temperature sensors is selected to optimize the understanding of the influence of the main heat sources of the machine on its thermal behavior.

EXPERIMENTAL SETUP

To confirm the proposed method, a set of measurements was performed in a large machine tool, in this particular case a column type milling machine with a total working volume of 10m x 6m x 1.4m. The machine measured is shown in figure 2.

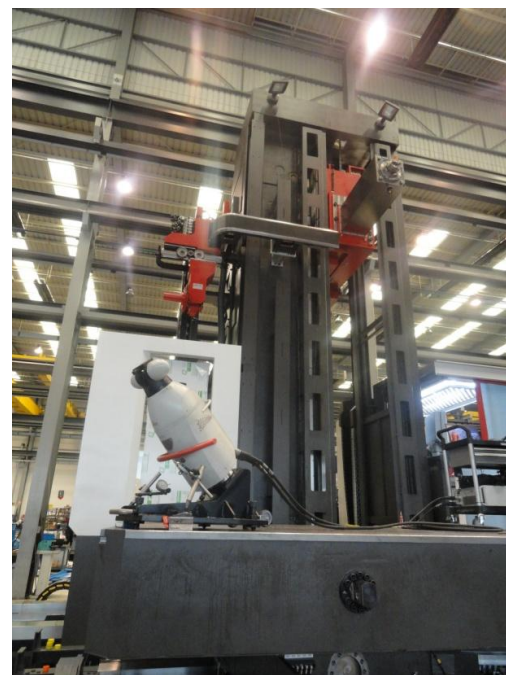


FIGURE 2. Measured machine.

Measured tooltip positions

The positional deviation of the column-ram ensemble was measured. Therefore it was not considered the position of the column along the longitudinal X axis. This also allowed getting a smaller measurement cadence for evaluating the thermal behavior of the machine over time. 12 points were measured, distributed in a plane of $X = 900\text{mm}$, $Y = 2400\text{mm}$, $Z = 1000\text{mm}$. The points in which the tooltip was positioned are shown in figure 3.

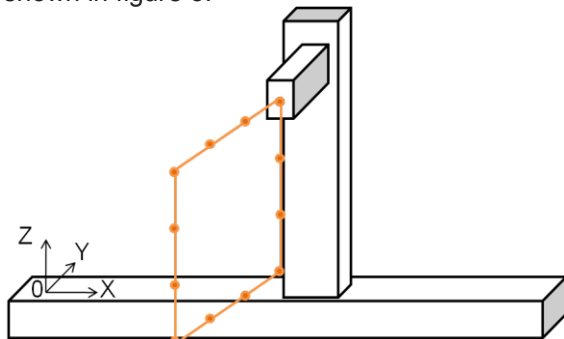


FIGURE 3. Measurement of tooltip positions.

Measurement cadence

Measurement cadence is directly influenced by the number of points measured in the machine work volume. Selected points made possible that a full cycle for measuring all the points from all the laser stations spanned 20 minutes. It is recognized that such a measuring cadence is sufficient when assessing the thermal characterization of big machine tools which have a long thermal inertia [7].

Location of laser stations

Optimal spatial positions were chosen to get the minimal measurement uncertainty, as shown:



FIGURE 4. Laser station locations.

Points for temperature measurement

Selected points for measuring temperature include the RAM (spindle, leadscrew nut, the front side and Y axis motor flange), column (top and bottom of the guides, top and bottom of the back side) and room air in several heights.

Test heating cycle

The machine heating cycle carried out was as follows: Initially two measurements were made with the spindle stopped (40mins). Afterwards, for machine warm up, two heating cycles were carried out (180mins), the first at medium speed and the second at full speed as shown in figure 8. Finally a cooling down was done. The main reason of doing such cycle was to deliberately obtain a rapid thermal distortion in the tooltip to make the measurements. It is understood that to carry out an adequate thermal characterization study it would be necessary to perform more heating and cool-downs cycles with different time periods [8].

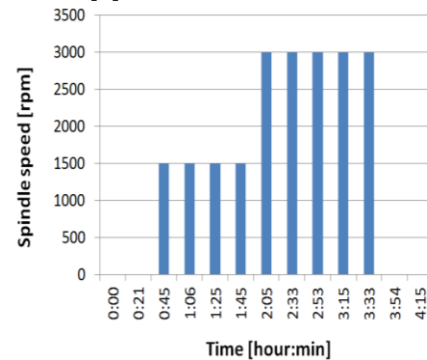


FIGURE 5. Thermal cycles.

EXPERIMENTAL RESULTS

In figure 6 some of the temperature measurement results are shown, in this case for the column and air temperature sensors.

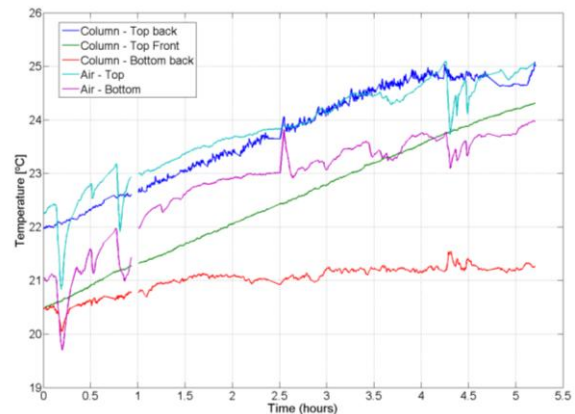


FIGURE 6. Temperature results – Column and air sensors.

In figure 7, some angular deviation results are shown for one of the levels (inclination of the X axis of the RAM). Each line represents a different cycle that started at a certain time shown in the legend. Within each line the inclination measurements of each of the 12 positioning points of that cycle are shown. It can be seen the angular deviation suffered by the machine during the test due to thermal effects.

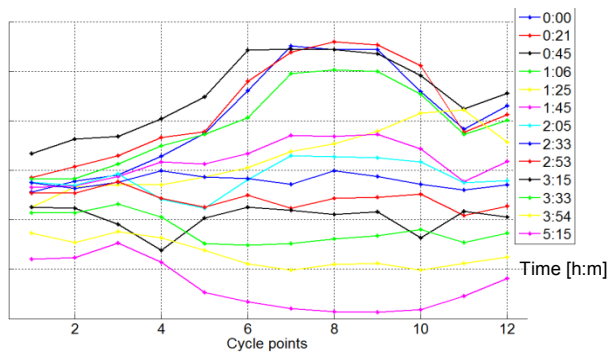


FIGURE 7. Inclination results (not at scale)

In figure 8 the multilateration measurement results for the tooltip volumetric positioning deviation is shown. The thermal distortion in each point is not scaled respect the scale of the positioning in space. A timestamp is shown for each of the cycles during which the machine was positioned in the 12 points.

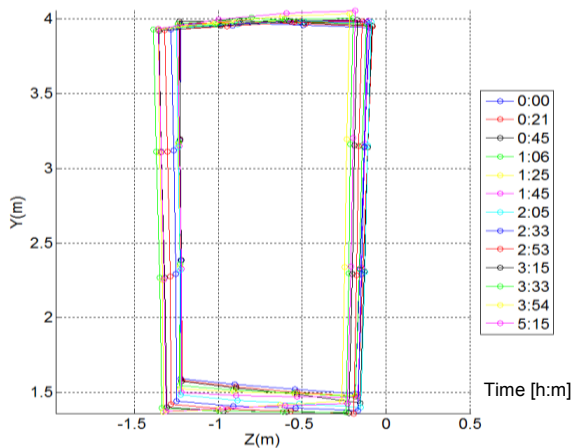


FIGURE 8. Tooltip thermal distortion (not at scale)

It can be seen how the rectangle composed by the 12 points in which the machine was cyclically positioned suffered a positioning deviation which changed over time. This thermal distortion was measured with a good continuity of the measured points and smooth tendencies.

FUTURE WORK

The next work will be focused in adapting this methodology to other machine tool configurations such as bridge-type machines with 1 degree of freedom translational table.

CONCLUSIONS

A new method for measuring the thermal distortion of large machine tools has been provided. An experimental measurement of the thermal distortion of a large column-type milling machine has also been presented, proving that the proposed method is suitable for obtaining this kind of measurements that can be used to perform the thermal characterization of the machine.

ACKNOWLEDGMENTS

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ORIENTATION CONTROL OF ANISOTROPIC PARTICLE BY SHEAR FORCE

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INTRODUCTION

The orientation of particles in ceramic compacts is an essential factor for controlling the performance and quality of ceramic compacts, such as a capacitor, a sensor, and a piezo element. Some ceramic particles have electrical orientations that depend on their crystal structure. When particles are arranged in the same direction, the electrical performance of ceramic compacts can be enhanced compared with that of ceramic compacts obtained without treatment [1]. We can change or improve the performance and quality of ceramic compacts by controlling the orientation of particles.

There are some techniques for controlling the orientation of particles: extrusion, pressure, tape casting, filtering, and slip casting methods. Anisotropic ceramic particles, such as plates and fibers, are often used in these methods. In the extrusion method, particles are oriented with their long side parallel to the extrusion direction. In the pressure method, particles are oriented with their long side vertical to the pressure direction. In the tape casting method, particles are oriented with their long side parallel to a substrate. All of these methods are not suitable for manufacturing a film or tube consisting of particles that are oriented with their long side parallel to the thickness direction (Figure 1). Unlike in these methods, in the filtering and slip casting methods, particles are oriented with their long side parallel to the plane to absorb water, however, the orientation degree is not uniform and depends on the location of a compact because the rate of sedimentation changes during the process [2]. If we can solve this problem, we can improve the performance of some solid electrolytes of cells or sensors and also produce new types of product, such as hollow fiber dialyzers and heat exchangers.

We have developed a new technique that enables us to continuously manufacture a ceramic tube consisting of particles that are

oriented with their long side parallel to the thickness direction. With a tube cut open, we can also obtain a film consisting of particles with their long side parallel to the thickness direction.

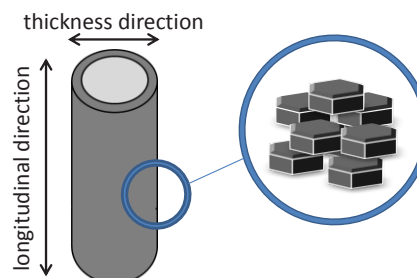


FIGURE 1. Explanation of thickness and longitudinal directions.

PRINCIPLE

An anisotropic particle (e.g., plate or ellipsoid) in shear flow is rotated by shear force. The rotation angle ϕ is described using Jeffery Orbits [3].

$$\phi(t) = \arctan \left(r \cdot \tan \frac{\dot{\gamma} \cdot t}{r + \frac{1}{r}} \right)$$

Here, t is time, r is the aspect ratio of a particle, and $\dot{\gamma}$ is the shear rate. This formula shows that the shear rate has no effect on the orientation degree but has an effect on the rotation speed. The rotating particle among the shear rate and interparticle interaction reached an equilibrium state. As a result, the particles on average appear oriented in the direction of shear flow [4]. The higher the aspect ratio of the particle becomes, the longer the particle stays oriented in the direction of shear flow (Figure 2). This indicates that the particle with a high aspect ratio is more likely to be oriented in the direction of shear flow. Some researchers mentioned and demonstrated that the orientation degree of an anisotropic particle in shear rate was affected by the particle size and volume fraction, which enhance the interparticle interaction. It was also mentioned that the shear rate and viscosity of slurry consisting of the particle and dispersion medium did not affect the orientation degree.

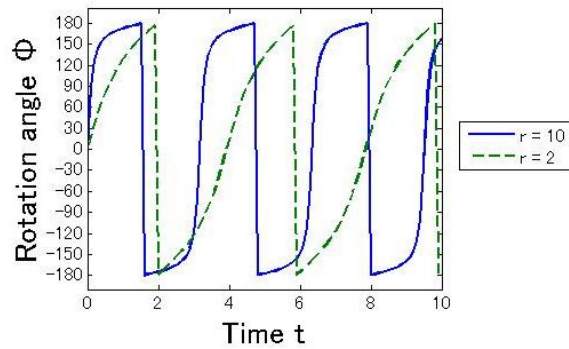


FIGURE 2. Rotation angle ϕ calculated using Jeffery Orbits as a function of time t where $\dot{\gamma}$ is 10. r values are 2 in the green dash line and 10 in the blue solid line.

There are numerous studies on the orientation of an isotropic particle using shear flow; however, most of these studies are about tape casting, which yields a compact that is several hundreds of micrometers in thickness. However, a much thicker compact has never been manufactured. Furthermore, the orientation degree depends on the probability of the orientation of each particle caused by the rotation of particles and interparticle interaction.

Focusing on both the particle rotation by shear flow and the interparticle interaction, we demonstrate a new way to more steadily manufacture larger compacts with a high orientation using shear flow.

EXPERIMENTAL RESULTS AND DISCUSSION

In this study, we use α - Al_2O_3 particles (KINSEI MATEC CO., LTD.), which were about 10 μm in diameter and 0.3 μm in thickness, and had a plate-like shape (Figure 3). Slurry was prepared by mixing them with an organic solvent binder. α - Al_2O_3 was about 23 vol%.

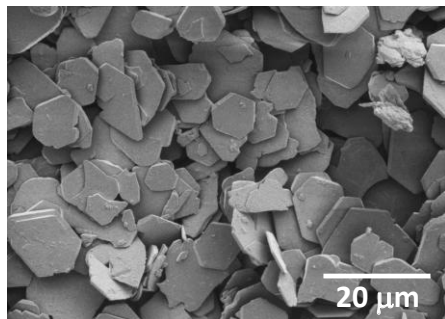


FIGURE 3. Scanning electron microscopy (SEM) image of α - Al_2O_3 particles.

Type Of Flow Field

We compared Couette and Hagen-Poiseuille flows in terms of the competence of the orientation of particles. Although the shear rate is uniform in Couette flow, it is not in Hagen-Poiseuille flow. The former is caused by the upper and lower basements sliding relatively. The latter is caused by a static basement and a drive force. In Couette flow, we obtained a highly oriented compact of about 100 μm thickness (Figure 4). In Hagen-Poiseuille flow, we obtained that of about 50 μm thickness. It is considered that the uniform shear rate in Couette flow aids the orientation at a larger volume; thus, we adopt it to form a flow field for further experimentation in this study.

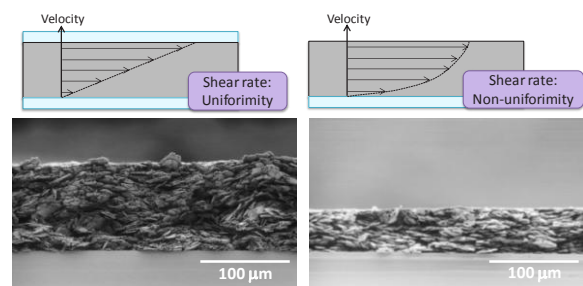


FIGURE 4. SEM images of cross sections of compacts.

(a) Compact by Couette flow.

(b) Compact by Hagen-Poiseuille flow.

Sedimentation In Shear Flow

We continued to induce Couette flow between the two basements and obtained a compact. Figure 5 shows our experimental system. A cylindrical container 50 mm in diameter was prepared and the upper basement was rotated coaxially at a constant speed of 40 rpm with a 5 mm gap filled with slurry. The shear rate was calculated to be approximately 20 per second within a 20-25 mm radius. After an approximately 5 hour treatment, we obtained a compact 2.5 mm in height. Compared with a compact without shear flow, that with shear flow was highly oriented at almost all places in the cross section. We defined the ratio of the number of particles per micrometer in the X-axis and to in the Y-axis as the orientation degree. The orientation degree of a compact obtained by adding shear force was about twice larger than that of a compact obtained with no treatment (Table 1).

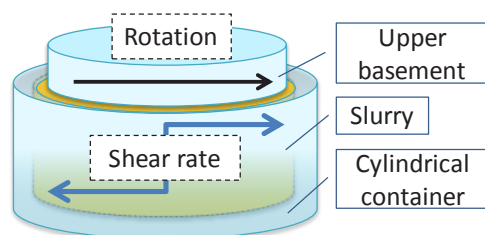


FIGURE 5. Schematic view of experimental system for orientation control.

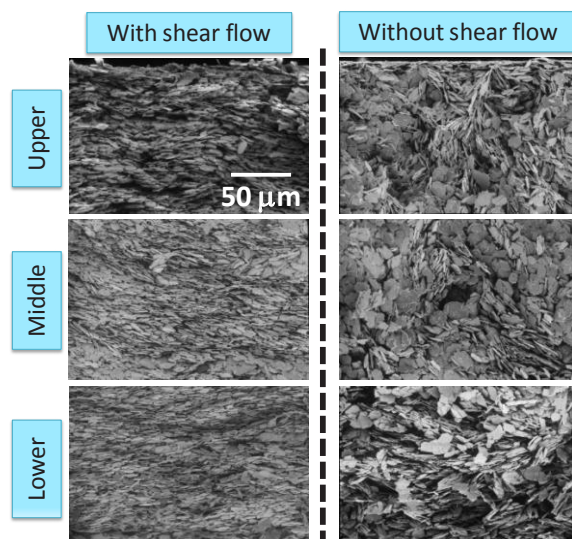


FIGURE 6. SEM images of cross sections of compacts.

TABLE 1. Orientation degrees of compacts.

	With shear flow	Without shear flow
Upper	2.97	1.18
Middle	2.89	1.22
Lower	3.68	2.07

It is clear that particles are oriented during the entire repeated process. The reason for this is shown in Figure 7. In shear flow, particles rotate continuously and become oriented to a certain extent owing to interparticle interaction. The particles also settle gradually onto the lower basement. When the rotating particles hit the lower basement, their rotation was suppressed. As a result, the particles completely settle parallel to the basement and they become a new basement. Repeating this process develops a large compact with a high orientation. Note that the oriented particles are fixed because the solvent is ejected owing to the decrease in gap between the particles.

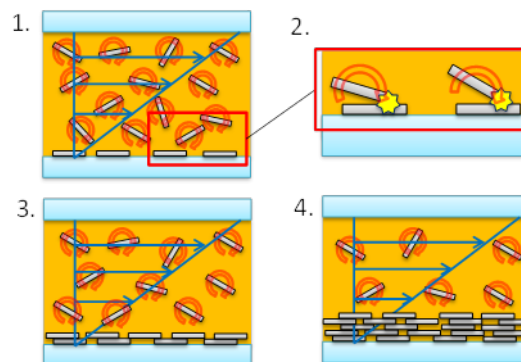


FIGURE 7. Schematic view of mechanism of orientation during sedimentation in shear flow.

Effect Of Volume Fraction

As we introduced, the volume fraction is considered to be an important factor for the orientation of particles. We prepared about 12 vol% slurry and conducted the same experiments as that with 23 vol% slurry. A highly oriented compact was also obtained, which showed a result almost the same as that with 23 vol% slurry (Figure 8). The obtained orientation degrees were almost the same (Table 2). This is considered to be due to the lack of difference in the volume fraction of particles in the slurry near the basement. When the particles settle onto the basement, the volume fraction near it is considered to reach the same value regardless of the original volume fraction in the slurry. We clarified that the volume fraction had little effect on the orientation degree in this method.

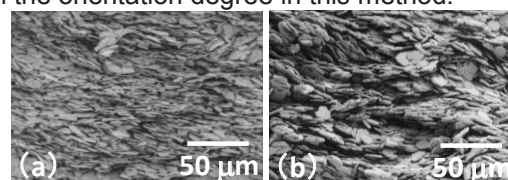


FIGURE 8. SEM images of cross sections of compacts.

(a) Compact with 23 vol% slurry.

(b) Compact with 12 vol% slurry.

TABLE 2. Orientation degrees of compacts.

	23 vol%	12 vol%
Middle	2.89	3.10

Effect Of Thickness Of Slurry

To determine the relationship between the orientation degree and the height of slurry, we conducted other experiments. We increased gap from 5 mm to 10 mm at a speed of 40 rpm,

resulting in the shear rate being reduced to half. Although it is considered that the shear rate has little effect on the orientation degree, we could not obtain an oriented compact (Figure 9 and Table 3). We also conducted an experiment at a speed of 80 rpm, which yielded the same shear rate as that of the experiment with a 5 mm gap. However, we still could not even obtain a compact, indicating that the particles did not settle at all. We calculated two shear rates. One was defined as $\dot{\gamma}_h$ in the thickness direction, and the other was defined as $\dot{\gamma}_r$ in the radius direction. We also defined the ratio of shear rates as χ .

$$\dot{\gamma}_h = \frac{2\pi r \omega}{h}, \dot{\gamma}_r = 2\pi \omega$$

$$\chi = \frac{\dot{\gamma}_h}{\dot{\gamma}_r} = \frac{r}{h}$$

χ shows the intension of the orientation force in a direction parallel to the basement, which we desire. Although χ ranges from 4.0 to 5.0 with 5 mm gap / 40 rpm, it ranges from 2.0 to 2.5 with 10 mm gap / 40 rpm and 10 mm gap / 80 rpm. It is considered that the orientation force in a desirable direction relatively decreases with an increase in the force in an undesirable direction, resulting in the random movement of particles. When the absolute amount of $\dot{\gamma}_r$ is large, the particles do not settle owing to their random movement in the slurry.

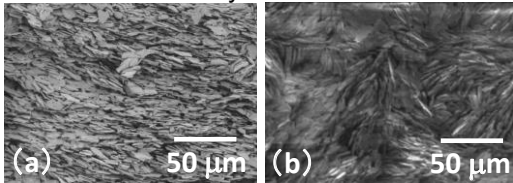


FIGURE 9. SEM images of cross sections of compacts.

(c) Compact with 5 mm gap / 40 rpm.

(d) Compact with 10 mm gap / 40 rpm.

TABLE 3. Orientation degrees of compacts.

	5 mm / 40 rpm	10 mm / 40 rpm
Middle	2.89	1.51

We also calculated the vorticity in the cylindrical coordinate system.

$$\omega_r = \frac{1}{r} \frac{\partial v_z}{\partial \theta} - \frac{\partial v_\theta}{\partial z}, \omega_\theta = \frac{\partial v_r}{\partial z} - \frac{\partial v_z}{\partial r}, \omega_z = \frac{\partial v_\theta}{\partial r} + \frac{v_\theta}{r} - \frac{1}{r} \frac{\partial v_r}{\partial \theta}$$

In our experiments, v_z and v_r were ignored; thus, the vorticities were calculated as follows.

$$\omega_r = \frac{\omega}{h} r, \omega_\theta = 0, \omega_z = \frac{2\omega}{h} z$$

We compared the vorticities of flow fields with 5 mm gap / 40 rpm and 10 mm gap / 80 rpm, which yield the same flow rate. The vorticities were the same; thus, we clarified that the reason for the absence of sedimentation with 10 mm gap / 80 rpm was not diffusion by flow.

Facilitated Sedimentation

The time required for the orientation process is so long that it is difficult to industrialize the process. If we can apply a larger gravity force to particles, they are considered to settle faster and the time required becomes shorter. We increased the gravity force by applying a centrifugal force. Sedimentation was completed within 5 s when an approximately 10^5 gravity acceleration was applied to the slurry using a centrifugal machine (MC-150, TOMY SEIKO CO., LTD.). Therefore, we could facilitate the sedimentation of particles by applying a large gravity acceleration.

CONCLUSION

We proposed a new technique for manufacturing highly oriented ceramics. We actually obtained a large compact about 2.5 mm in thickness consisting of particles with a high orientation degree. In this method, the volume fraction of particles in the slurry had little effect. The intensity of shear rate in the thickness direction compared with that in the radius direction was considered to have a large effect on the orientation degree of a compact.

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Design of Ignition Tuning Targets for Indirect Drive Ignition Experiments on NIF

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The National Ignition Campaign (NIC) began ignition experiments on NIF in FY2010. Ignition tuning targets generate physics information on target conditions to help scientists refine ignition experiment parameters. Three of the tuning targets currently fielded include “keyhole” targets that provide information on shock timing and velocity, “re-emit” targets that facilitate study of the capsule implosion shape and “convergent ablator” targets that help measure implosion velocity and capsule mass at the end of the implosion. We will discuss the tuning targets requirements and how these have been met with current target design, fabrication, assembly and metrology.

LLNL-ABS-503491

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A SHOP AIR SUPPLY FOR A HUNDRED SPINDLES

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The problem

Wet air has been the cause of many preventable air bearing failures and by describing our in-house system we hope to spare other users this kind of trouble. Internal corrosion (Fig 1) is an unnecessary type of repair, and the most expensive, especially since it often involves multiple units. Repairing a rusty spindle can cost as much as 2/3 the price of a new spindle so a reliable air system is cost-effective.



FIGURE 1. As air expands it dries and here we see rust at the start of the thrust bearing flow path. This 4R cost \$5,000 new and \$3,500 to repair

The system

Since nothing is perfect, even a top quality dryer is subject to failure. Our in-house system has a second air dryer in case of a problem with the main dryer. We also have manual drains after the automatic drains. Activated carbon canisters are used as final filters at each spindle to catch any oil or particulates that may have entered the piping system downstream from the primary system.

The crucial requirement is that water must never condense in the spindle. Since some spindles are in machines with coolant, we leave them on air at all times, but when a spindle is not in use, 40 psi is sufficient to keep liquids from wicking in.



Figure 2. This variable-speed 67 HP, air-cooled rotary-screw air compressor produces up to 200 SCFM at 185 PSI. A heat exchanger reduces the outlet air from ~200°F to ~100°F. Waste heat augments the HVAC when needed.

After the air is compressed (Fig 2), it is dripping with particle-laden oily water and must be filtered to avoid clogging the refrigerant dryer (Fig 3). Most of the liquid is stripped by the first filter and drained automatically. Transparent drain lines enable visual inspection to assure that automatic drains are functioning. Manual valves are used in various locations where only occasional draining is required. All drain lines are combined in a PVC manifold which runs to an oil-water separator.



Figure 3. This refrigerant dryer cools 100°F air to 38°F. It cost \$4,604 and the filters cost \$315 each. A second pair of filters remove liquid which may have carried over. They also allow the air to warm a bit before it enters the receiver. Inset: Liquid water is drained automatically at the coldest point.



Figure 4. This \$1,679 receiver contains 400 gallons of clean, dry air, assuming the dryer,

filters, and drains are fully functional.



Figure 5. In case they fail, we are backed up by a \$2,100 passive deliquescent dryer which does not require electrical power and in theory should never come into play.

It is in a separate room from the air compressor so that it is not too warm.

Dew point depression is about 20°F, which is adequate for our purposes, assuming a spindle is not significantly cooler than the air in the dryer.

In the case of a grinding spindle cooled by evaporation from spray-mist coolant, then additional drying is required such as drying by expansion at reduced PSI. (Note that both dryers operate at full line pressure.)

An activated carbon filter (\$370) removes residual oil vapor and protects pressure regulators from particulate contamination.



Figure 6. 140 PSI air goes to the shop (left) and 165 PSI goes to the lab. During the day the yellow-handled valves are open and at night they are closed so that the air flows through 40 PSI regulators. Two-inch diameter pipes are used to minimize pressure drop.



Figure 8. Regulators maintain the desired pressure at each point of use.



Figure 7 This oil/water separator processes the drain liquids so that the oil can be sent for recycling.



FIGURE9. An electronic humidity indicator can be equipped with an alarm and a relay for switching to an alternate air supply. The optimum placement is a high-pressure line feeding a spindle in a cool environment.



FIGURE 10. The last line of defense is a filter with activated carbon having enough capacity to hold back a wet air event for several hours.

Conclusion

An air system for air bearing spindles must never allow condensation on bearing surfaces. What seems like excessive redundancy is cost-effective insurance against a wet air catastrophe.

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AUTOMATIC CENTERING OF PRECISION TURNED PARTS ON A VACUUM CHUCK

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INTRODUCTION

One of the first steps in many machining processes involves aligning the workpiece with axes of the machine. For a lathe, this involves aligning the workpiece with the center of rotation of the spindle. If done manually, this centering procedure can be laborious, time-intensive, and error-prone. The goal of this research is to develop a device to be used inside a machine enclosure that will align the center of a part to within $2.5\text{ }\mu\text{m}$ of the spindle center automatically in less than 3 minutes.

Previous research by Furst [1] showed that a part could be precisely centered using a compact setup featuring a voice coil actuator. The research outlined here involves a more automated solution to part centering called Thumper II. Thumper II operates while the machine spindle continuously rotates at a known constant speed less than 20 rpm. The Thumper II alignment system operates without connection to the machine tool axes and uses the constant spindle speed to estimate angular position of the runout signal.

THUMPER II DEVICE

The Thumper II device is able to measure part runout, as well as precisely tap the part to move it towards the center of rotation. The device works in 2 modes: measurement and tap. The internal components are shown in Figure 1.

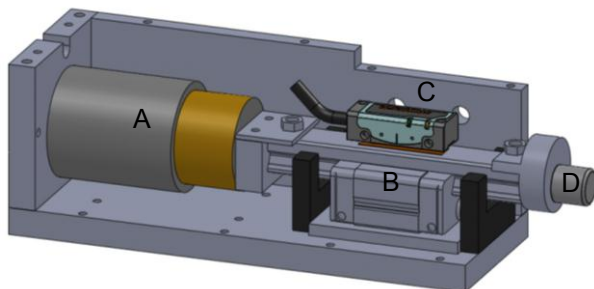


FIGURE 1. Construction of Thumper II.

Measurement Mode: In measurement mode, the voice-coil actuator (A) supplies a light pre-load force such that the tip (D) contacts the part.

As the part rotates, the encoder (C) reads the displacement of the tip, which is a measurement of part runout, as a function of time. Assuming the speed of the spindle remains constant, the time can be turned into spindle position. The resulting displacement vs. spindle position data can then be used in a least-squares circle fit to find the center of the part and thus the location of the high spot.

Tap Mode: In tap mode, a variable voltage signal with a duration of 20 ms is applied to the voice-coil actuator. This signal accelerates the coil-rail-tip assembly to a desired impact velocity. The assembly will collide with the part at or near the high spot and move it towards the center of rotation.

The Thumper II device can be seen in Figure 2 with a protective bellows and clear plastic shields. The dimensions of the aluminum structure are: 60 x 50 x 150 mm (2.3" X 2.0" X 6.1") and it weighs 1.4 Kg (3 lbs).

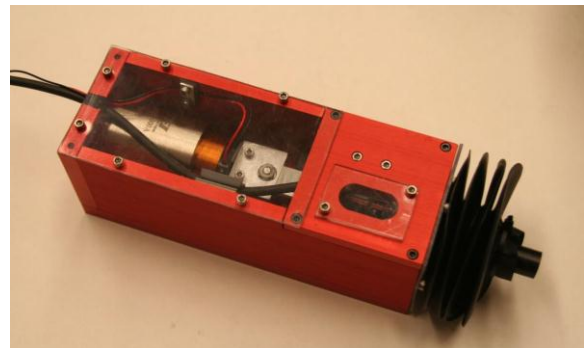


FIGURE 2. Thumper II device.

THUMPER II CONTROL SYSTEM

The control system for Thumper II features a laptop for the user interface, a microcontroller to calculate the amplitude and location of the tap and an amplifier/power supply to drive the voice coil. The microcontroller includes on-board quad decoder chips to read the optical encoder. The analog output of the microcontroller sends precisely timed analog pulses to the amplifier. The microcontroller runs an algorithm that determines when and how hard to tap the part,

as well as the circle fit algorithm that a least squares circle to the collected data. The center of this circle corresponds to the angular location of the high spot.

A graphical user interface on a laptop is able to display information such as runout on the screen, as well as allow the operator to give the microcontroller information, such as part weight, which is used by the tap algorithm. The screen during alignment is shown in Figure 3.

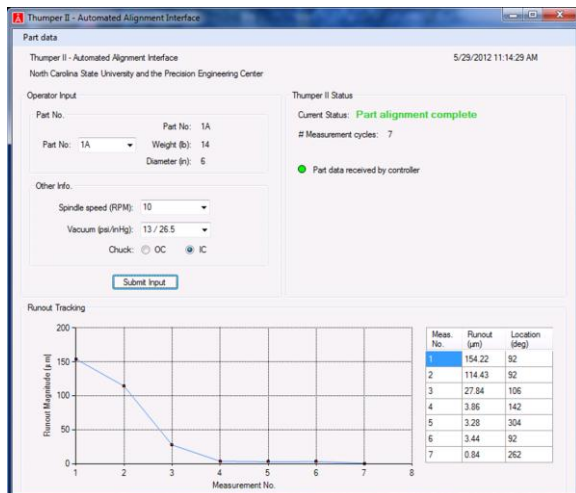


FIGURE 3. Graphical interface allows the user to input part data, spindle speed and provides feedback on the progress of the alignment.

ALIGNMENT PROCESS CONSTRAINTS

There are many challenges to automate the centering process as well as geometric constraints:

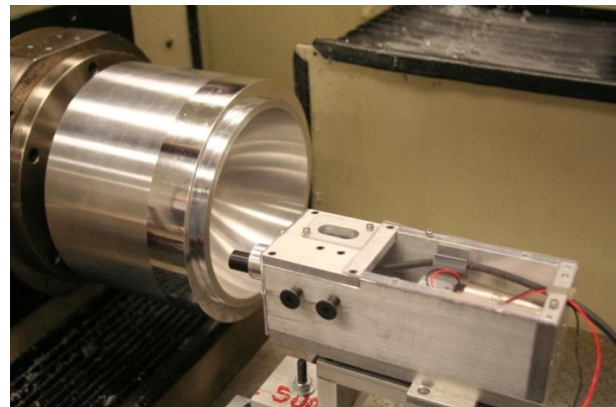
- **Thumper II Orientation:** The centering device was designed to replace a cutting tool on a machining center. As a result, it is oriented to tap the part horizontally, either in a radial or axial direction as shown in Figure 3.
- **Autonomous Operation:** The centering process must be accomplished without input from the machine tool controller.
- **2D part motion:** On a spindle with a horizontal axis, (Figure 4), the part will move horizontally due to the impact by Thumper II. However, it also falls vertically due to gravity. Understanding this 2D motion is vital to successful alignment.
- **Tap algorithm:** Given parameters such as part weight, initial misalignment and vacuum pressure, this algorithm must select the

initial tap magnitude and subsequent taps to quickly center the part.

- **Device sensitivity:** The device must be sensitive enough to ensure alignment of 2.2 Kg (5 lb) parts but also powerful enough to rapidly align a 16 Kg (35 lb) part with an initial misalignment of 1.25 mm.



a) Thumper II centering the outside of a shell



b) Thumper II centering the inside of a shell

FIGURE 4. Thumper II installed on DTM for alignment of a hemispherical shell.

ALIGNMENT PROCESS

The alignment process begins with a measurement of the location of the part center followed by an impact of standard magnitude at the high point. Based on the motion of the part center measured after this impact, a new impact force is calculated and implemented.

Alignment Error Measurement: The first step in the process is to find the initial alignment of the part with respect to the spindle rotation axis. Figure 5 illustrates a series of measurements of

the location of surface points in terms of radius and angle. The radius is the encoder reading of the contact point between the plastic tip and the part plus some nominal radius. The load and therefore wear of the nylon tip is negligible. The angle is calculated from the spindle speed input by the operator and the system clock for each point. A least-squares algorithm [2] is used to find the center of the circle and thus the misalignment.

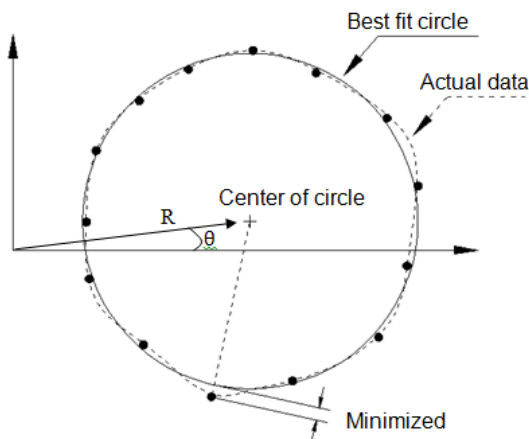


FIGURE 5. Least-squares circle fit method

Impact Dynamics: Based on the magnitude of the misalignment, the velocity of the impacting mass can be determined. The motion of the part depends on the friction between the part and vacuum chuck, the vacuum pressure and the mass of the part. The part displacement can be measured and related to the velocity of the Thumper II mass at impact. Figure 5 shows that displacement for a specific part is a function of the square of the impact speed or the momentum of the impacting mass.

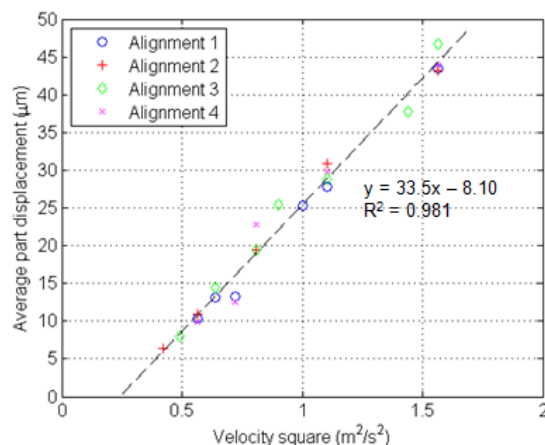


FIGURE 6. Relationship between impact velocity and part motion (30 lb part with 11.3 psi vacuum)

Impact Speed Generation: The Thumper actuation is accomplished in two steps: the first step is to apply a high current to accelerate the mass to the desired speed followed by a low current to overcome the drag in the ball bearing slide. This is illustrated in Figure 7 where the Thumper II mass is accelerated to 0.5 m/s in 20 ms using 1.2 amps of current.

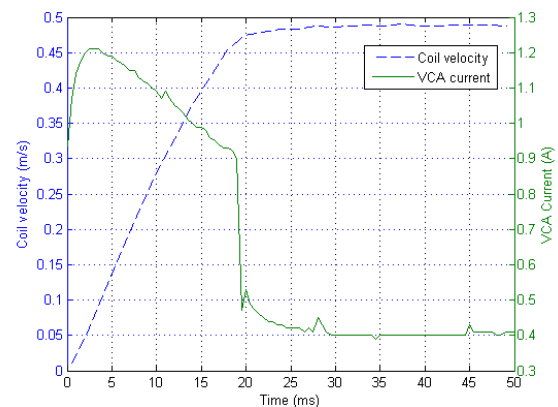
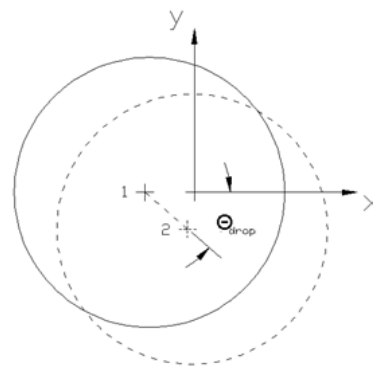


FIGURE 7. Actuation of the voice coil motor to accelerate the Thumper II mass.

Multiple impacts: Thumper II is able to tap the part near the high spot up to 3 times in one revolution. Tapping multiple times per revolution allows Thumper II to align parts more quickly. If it is observed that the maximum impact velocity of 1.25 m/s yields a part displacement less than the new misalignment value, it may be useful to tap the part multiple times during one revolution at 1.25 m/s.



FIGURE

Figure 8. Definition of the drop angle for a horizontal impact.

Drop angle: Figure 8 describes the horizontal and vertical motion of the part when it is tapped from the side. If the part is tapped at the high point each time and the drop angle is on the order of 60°, the part will move over and down

but stay at nearly the same misalignment. To overcome this problem, the part is hit after the high spot passes which will drive it toward the spindle center. The motion of the part from the first impact is used to find the drop angle and apply this correction to center the part.

RESULTS FROM THUMPER II TESTING

The fully-automated program has been tested for a variety of part weights, vacuum pressure, OC and IC chuck configurations and materials [3]. The set-up for these experiments is shown in Figure 4 for radial and axial alignment. The results of several tests are illustrated in Figures 9 and 10. These Figures show the locus of the part centers as the alignment proceeds. The location of the center is shown on a polar plot as an angular location and a radius. The radius is a log scale.

Drop Angle Compensation: For heavy parts with low vacuum pressure, it is not possible to align a part to $2.5\ \mu\text{m}$ by simply tapping the high spot. In this case, it is necessary to tap the workpiece when the high spot is above the Thumper II tip, such that the part will fall towards the rotational center as well as displace horizontally.

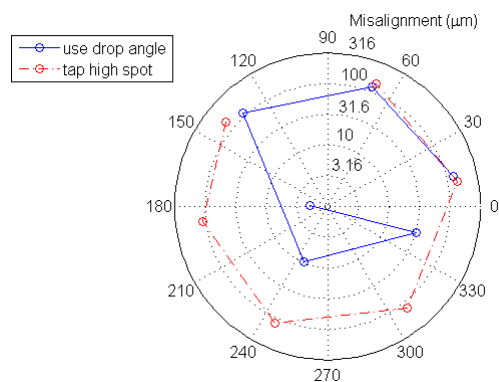


FIGURE 9. Centering of heavy (30 lb) part with and without drop angle compensation. Testing at 8 rpm on an OC chuck at 10.2 psi vacuum.

As seen in Figure 9, the workpiece could not be aligned by simply tapping the high spot. In five cycles of tapping only at the high spot, the workpiece misalignment remained around $100\ \mu\text{m}$. However, with five taps using drop angle compensation, the workpiece was aligned to $1.97\ \mu\text{m}$. It is important to note that the first three measurements are about the same ($\sim 100\ \mu\text{m}$) using compensation because for the first and second tap, drop angle values were calculated and used in the following taps.

Part Weight: Figure 10 shows a plot of the alignment process at a vacuum of 11.1 psi on the OC chuck for two parts of different weight. In both cases, the workpiece begins at a misalignment of $150\ \mu\text{m}$ and ends with a final misalignment of $3\ \mu\text{m}$. The 1 lb part displaces back and forth until the appropriate tap velocity is determined. The 30 lb part, however, follows a less direct path towards the center of rotation due to a larger drop angle. Thus, Figure 10 demonstrates the additional challenge of centering a heavy part on a vertical chuck face.

One interesting observation from Figure 10 is that the alignment process generally spends a significant number of iterations around the target goal. This is to be expected as variations in friction and displacement direction make it challenging to precisely position a workpiece. It should be realized that the angle drop capability was turned off in Figure 9 so the effect of part weight on the alignment process could be better observed.

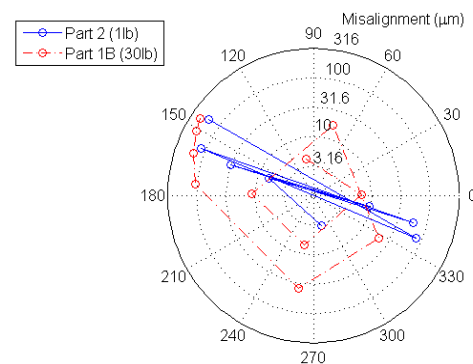


FIGURE 10. Automatically centering 2 parts with weights of 1 lb and 30 lbs at 11.1 psi vacuum.

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The Design, Fabrication, and Assembly of Material Equation-of-State Targets for the National Ignition Facility

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Abstract

The unique capabilities available at the National Ignition Facility (NIF) allow users to recreate the pressure-temperature conditions within the cores of giant planets for materials such as Carbon and Iron. The Equation-of-State (EOS) of these materials can be measured using ramp compression experiments. These experiments require use of high-fidelity targets as shown in Figure 1 and sophisticated diagnostics to acquire the ramp compression data.

When shot in the NIF, the high-fidelity EOS targets convert laser energy from the NIF into x-ray energy via gold hohlraums (essentially, thin walled cylinders with partial openings at each end). The X-ray energy then strikes the backside of an ablative material (the ablator) to create a shock that will be transmitted to the material to be studied. As the shock transfers through the material of interest the velocity of the front-side of the material is measured via a diagnostic that is part of the NIF (see Figure 2). The EOS can be determined from the velocity measurements of the free surface. Therefore, the two critical components of an EOS target are the hohlraum and the experimental package which will typically contain the ablator, shielding, and the material to be studied (e.g., carbon or iron).

In order to minimize negative effects on the velocity measurements caused by the deposition of large amounts of laser energy into the target, the target hohlraums are filled with 0.2 atm of gas; the gas slows down the formation of an undesirable shock. To contain the gas, the hohlraums must be sealed. The hohlraums are filled with gas while installed in the NIF target chamber (a vacuum environment) prior to shot time.

Not all of the laser energy directed at the target is desirable. A large portion of the laser energy is undesirable unconverted light that can

damage the experiment and possibly damage the laser facility via reflections off of the target. To mitigate this concern, extensive shielding is incorporated into the target to protect the target and scatter the undesirable laser energy away from critical hardware.

The target is held in the center of the target chamber via a 5-meter long target positioning arm. To simplify mounting of the target to the positioning arm, a universal kinematic mount is included as part of the overall target; it is attached to the hohlraum via a carbon rod. To ensure that the target is properly located in the target chamber, additional features (target positioning features – in this case, alignment fibers) are added to the target to allow the facility users to visually verify via multiple cameras that the target is positioned properly.

Designing and making the components just described and assembling these components into a functional EOS target will be the focus of this paper. Three of the significant challenges include making the experimental packages with stepped features out of carbon and iron, measuring the thickness of the stepped features, and assembling the fragile target components into usable targets.

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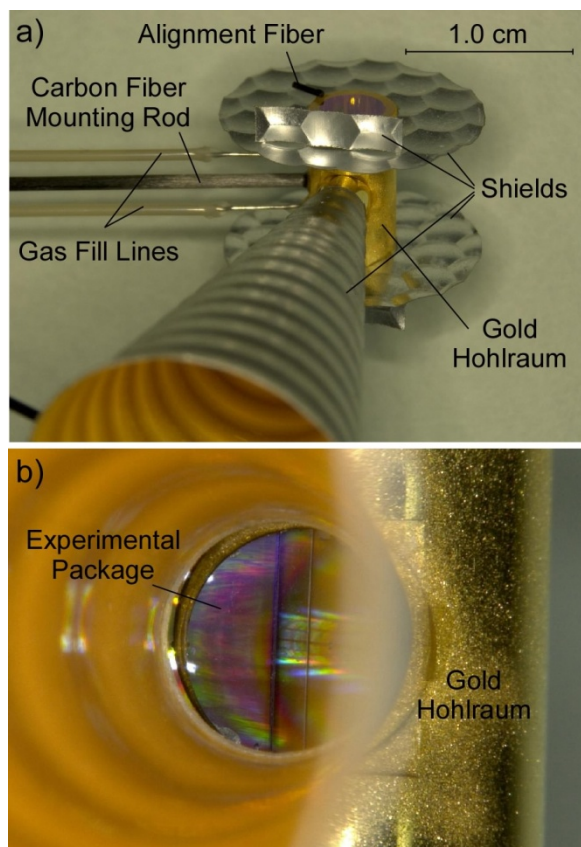


FIGURE 1. a) A NIF EOS Target. b) A view down the cone shield showing the experimental package that is attached and sealed to the gold hohlraum.

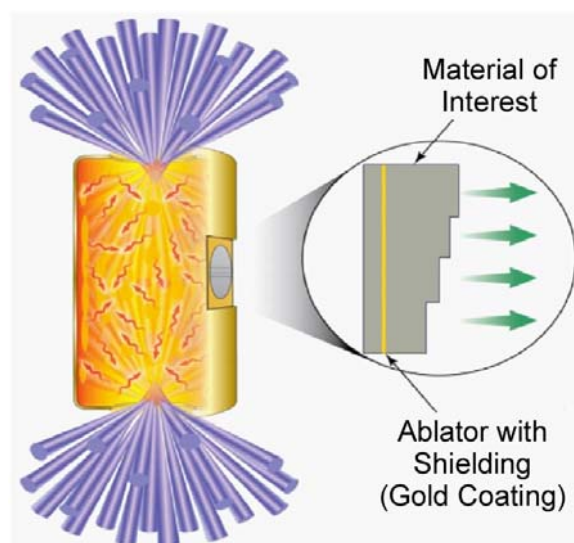


FIGURE 2. Illustration showing laser energy entering and striking the inside surfaces of the gold hohlraum, which produces x-rays that shock the experimental package.

Novel Approach for Slicing of Silicon Carbide by Multi-wire EDM using Wire Electrode with Track-shaped Section

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INTRODUCTION

Today, the reduction of greenhouse gas is one of important issues all over the world. The efficient use of electricity has become the very important task in order to continue the sustainable development of society. Silicon carbide is excellent in high insulation breakdown, large band gap energy and high thermal conductivity. The high performance semiconductor devices of silicon carbide with the high-speed responses could be expected under the high temperature condition. Moreover, it is possible to realize the size reduction and a highly efficient semi-conductor device. Therefore, silicon carbide has drawn attention as a material used for power devices in the next generation [1]. In order to reduce the cost and stabilize the supply of silicon carbide, the high efficient slicing method with a small kerf width is required. A multi-wire saw slicing method has been conventionally applied to the slicing of ingot. Since silicon carbide is hard and brittle, the combination of fixed abrasive diamond wire and high load is required for the efficient slicing of ingot [2]. Thus, this method has some problems such as cracks, large kerf width and so on.

On the other hand, the electrical discharge slicing method would solve these problems, and could perform the high-performance process for the silicon ingot [3, 4]. In addition, this method would be eco-friendly processing, because only deionized water is used as a working fluid. A multi-slicing technique with a single wire and a multi-EDM power supply unit was newly developed in order to increase the efficiency of wafer slicing [5, 6], since the number of sliced wafer was only one per one process by the conventional wire-EDM method. In the electrical discharge slicing, cracks generated on machined surface and kerf width decrease by

controlling the machining condition with a fine wire electrode, since the machining force is very small due to the noncontact processing.

In order to perform the more efficient and cost-effective slicing process, much higher wire tension would be required. In general, thin wire electrode with round section is normally incompatible with high quality kerf shape, since the wire tension must become smaller with decreasing the sectional area of wire electrode.

From the viewpoints mentioned above, electrical discharge slicing method of silicon carbide by a wire electrode with track-shaped section was newly proposed to realize much higher wire tension and small kerf width, and its possibilities were experimentally investigated.

EXPERIMENTAL PROCEDURES

Figure 1 is the photograph of multi-wire EDM slicing equipment consists of wire driving part and EDM power supply unit. Figure 2 schematically shows a wire running part of multi-wire EDM slicing equipment. The wire electrode was wound spirally on wire guides 40 times along the wire guide pitch. The wire electrode



FIGURE 1. Multi-wire EDM slicing equipment.

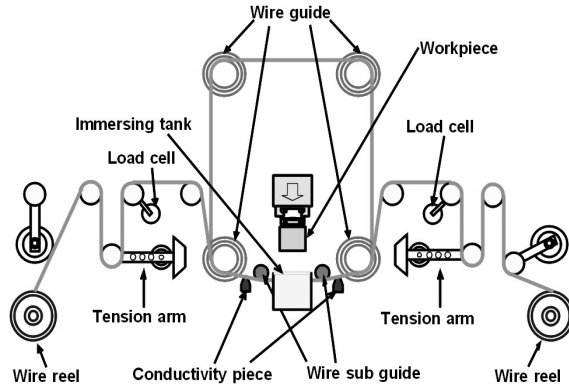


FIGURE 2. Schematic diagram of wire driving part in multi-wire EDM slicing equipment.

TABLE 1. Machining conditions.

Workpiece	Silicon carbide $\rho_s = 0.5 [\Omega \cdot \text{cm}]$
Electrode material	Brass coated steel
Wire tension	$W_T = 20-30 [\text{N}]$
Polarity	Electrode [-]
Wire speed	$W_s = 100-200 [\text{m/min}]$
Open circuit voltage	$ui_H = 283 - 300 [\text{V}]$ $ui_L = 120 [\text{V}]$
Discharge current	$i_e = 8 [\text{A}]$
Discharge duration	$t_e = 1 [\mu\text{s}]$
Duty factor	$\tau = 14\%$
Dielectric working fluid	Deionized water $\rho_w = 25 [\text{k} \Omega \cdot \text{cm}]$ $\theta_f = 20 [^\circ\text{C}]$

was fed reciprocally by reversing the feed direction, when the wire electrode came to the end of set travel distance. The silicon carbide ingot was sliced by moving the workpiece in the downward direction. The wire sub guides were prepared at the nearest side to workpiece in order to stabilize the wire position. Three sets of conductivity piece for supplying discharge pulse to the wire electrode were placed at the position between wire guides and wire sub guides. Maximum number of machinable wire electrode is limited to three according to the number of the multi-power supply unit, which can generate three individual discharge pulses at the same time.

Table 1 shows main machining conditions used in this study. A multi-EDM power supply unit can generate lower and longer discharge pulse compared with the normal commercial wire-EDM equipment. This electric discharge power supply has a high voltage circuit to start the electrical

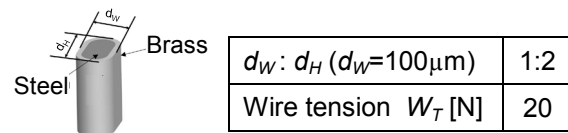


FIGURE 3. Schematic illustration of wire electrode with track-shaped section.

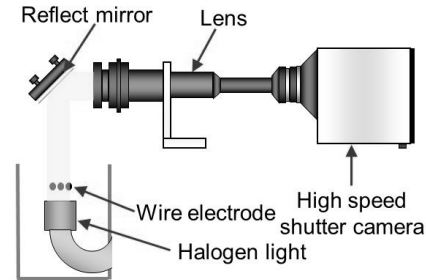


FIGURE 4. Observation method of wire vibration by high-speed shutter camera.

discharge and a low voltage circuit to keep the electrical discharge. In this study, an open circuit voltage in a high and a low voltage circuit were named as ui_H and ui_L respectively. Silicon carbide of $0.5 \Omega \cdot \text{cm}$ in a specific resistance was used as a workpiece, and the width for slicing was 80mm. The wire electrode was set as negative polarity. A deionized water of 20°C was used as a dielectric working fluid, and its specific resistance was set to $25 \text{k} \Omega \cdot \text{cm}$.

A brass coated steel wire electrode with track-shaped section was used as shown in Fig. 3. This wire electrode consists of a steel core wire and a brass coating of $0.18 \mu\text{m}$ on the core wire [6]. The width d_W and height d_H of wire electrode were $100 \mu\text{m}$ and $200 \mu\text{m}$, respectively. The height direction of wire electrode was arranged in parallel to the machining direction.

The vibration of wire electrode was observed by a high-speed shutter camera under the only wire running condition without supplying electrical discharge pulses as shown in Fig. 4. The light source was a halogen lamp, and the shadow of wire electrode was recoded as the vibration.

Figure 5 shows observation results of non-vibrated and vibrated wire electrode. The base is the original width d_W of wire electrode with track-shaped section, and d_m is the width of tilted and moved wire electrode. The vibration amplitude A_W in one side of wire electrode was calculated by the equation (1).

$$A_W = \frac{d_W}{2} \times \left(\frac{d_m}{d_W} - 1 \right) \quad (1)$$

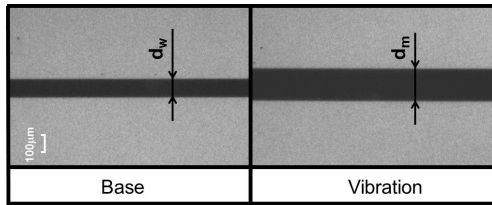


FIGURE 5. Example of wire movement observation by high-speed shutter camera.

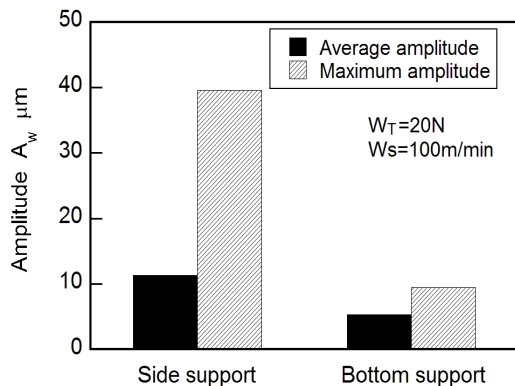


FIGURE 6. Amplitude of wire electrode vibration for track-shaped section by both bottom and side supporting methods.

EXPERIMENTAL RESULTS AND DISCUSSION

Investigation of Stable Wire Running

When the wire EDM slicing of silicon carbide was carried out by using brass coated steel wire electrode with track-shaped section, the kerf width became more than twice of 100μm in width of wire electrode with track-shaped section. It is considered that the slicing was done without the correct orientation of wire width 100μm to the machining direction, and the incorrect orientation of wire width would decrease the processing efficiency and quality even under the higher wire tension. Therefore, the supporting method of wire electrode near the processing point was investigated. In the multi-wire EDM slicing method, the bottom supporting method, in which the damper pressed the bottom of wire electrode in upward direction, had been used to decrease the vibration of wire electrode with round section. However, further prevention of wire movement was required, and the reduction of wire vibration was challenged by the side supporting method, in which both sides of wire electrode in the width direction were held by fixture plates.

In order to investigate the effect of proposed side supporting method, the vibration amplitude in one side of wire electrode was measured by

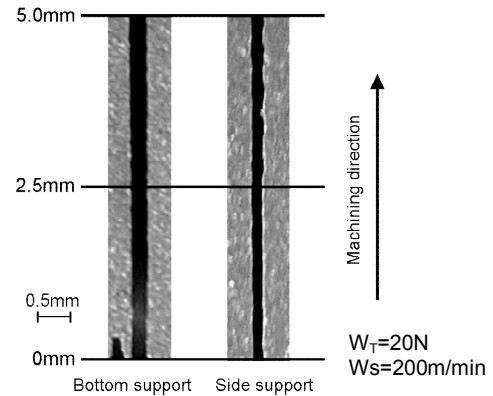


FIGURE 7. Kerf shapes by wire electrode with track-shaped section by both bottom and side supporting methods.

both supporting methods as shown in Fig. 6. The side supporting method could decrease the vibration amplitude of wire electrode significantly. Both maximum and average amplitude could be improved, and the standard deviation of side supporting method also became smaller than that of bottom one.

Figure 7 shows the kerf shapes of silicon carbide by single wire slicing using the brass coated steel wire electrode with track-shaped section for both bottom and side supporting methods at wire tension 20N. Although the side supporting method remained some irregularities in the kerf shape, the smaller kerf width could be obtained compared with the bottom supporting method. The removal rate with the side supporting method was 5.1mm²/min, which was higher than 4.5mm²/min with the bottom supporting method. It was cleared that the side supporting method made it possible to keep the stable wire running and reduce the kerf width even in the case of wire electrode with track-shaped section.

Influence of Wire Tension

As shown in Figure 7, although high quality kerf could be obtained by the side supporting method, there remained some irregularities in the kerf shape. In order to improve the quality of kerf shape, much higher wire tension of 26N, 28N and 30N was loaded, and the slicing of silicon carbide was investigated.

Figure 8 shows the kerf shapes for various wire tension by single wire electrode with track-shaped section in the case of side supporting method. As shown in the figure, the irregularities in the kerf shape became unremarkable with increasing the wire tension, since the stable wire

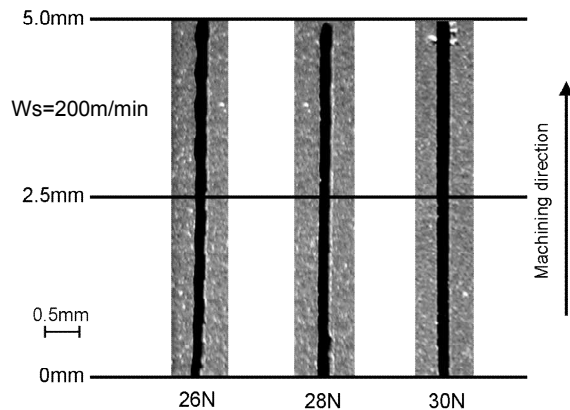


FIGURE 8. Kerf shapes for various wire tension by wire electrode with track-shaped section in the case of side supporting method.

TABLE 2. Kerf width for various wire tension by wire electrode with track-shaped section in the case of side support method.

Wire tension T_N [N]	26	28	30
Min. kerf width [μm]	152	155	166
Max. kerf width [μm]	209	249	188

running could be performed under the higher wire tension condition.

Table 2 shows the measurement results of kerf width for various wire tension by single wire electrode with track-shaped section in the case of side supporting method. Since the difference of kerf width between minimum and maximum value became the smallest by wire tension of 30N, the amplitude of wire electrode with track-shaped section could be improved and kept at the smaller level to obtain straight kerf shape.

As mentioned above, the combination of side supporting method and higher wire tension could lead to the stable processing result with uniform and straight kerf shape, and it was cleared that this proposed method by wire electrode with track-shaped section had the possibility of high performance slicing. However, the gap distance between wire electrode and workpiece was still larger, and the further stabilization of wire running should be investigated in order to gain the high ability of this proposed method by wire electrode with track-shaped section.

CONCLUSIONS

Electrical discharge slicing method of silicon carbide by a wire electrode with track-shaped section was newly proposed to satisfy both high

quality and narrow kerf, and its possibilities were experimentally investigated. Main conclusions obtained in this study are as follows.

- (1) The side support method with fixture plates could improve the running stability of brass coated steel wire electrode with track-shaped section and reduce the kerf width.
- (2) The brass coated steel wire electrode with track-shaped section could accomplish the uniform and straight kerf shape by increasing the wire tension.

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DIGITAL LITHOGRAPHY MASKING METHOD ON FABRICATION OF PRECISE UNIFORM MICRO-FEATURE ARRAY WITH 16 μm RESOLUTION

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INTRODUCTION

Current demands in various industries such as electronics, optics, aerospace, and biotechnology need manufacturing process that can fabricate micro-parts. Micro-feature fabrication requires reliable and repeatable methods [1]

Protein microarrays constitute a significant tool for studying biomolecular. One of it type is microwell chip. They consist of spatially arranged protein spots immobilized usually on planar surfaces. The most common industrially accessible fabrication method of protein microarrays is based on attachment of proteins on chemically derivatized glass slides [2], where protein solutions are delivered by means of high-precision printing robots [3].

Superhydrophobic surface, characterized by water contact angles (CA) greater than 150° , have been attractive for various applications [4], [5]. Specifically, water repellence is useful dust-free coating, a covering that prevents snow from sticking to antennas and windows, and reduces the friction of water flow. Many plants, including the lotus leaf, naturally exhibit the unusual wetting with this kind of rough architecture. In particular, the surface can be consists of micron-sized pillar.

Above two cases are example applications of the micro-feature array.

In the other hand, we have biomachining process as a machining process that uses microorganisms to remove metal from a workpiece. This process basically utilized biological process of microorganism to cut the workpiece of metal [6]. Microorganism we used is bacteria such as *A. ferrooxidans* and *A. Thiooxidans*. Thus this process can be categorized as micromachining thank to the bacteria size of about 1 μm .

As the techniques for micro-fabrication, both of them are usually combined with a patterned mask.

UV-based maskless digital lithography is an affordable candidate of this masking, since the projected image is formed by a multitude of UV beamlets, which overcomes the shortcoming of lower production capacity and high investment of laser or electron beam, and it does not require the use of a rigid mask, which replace by a computer assisted micromirror array [7]. The "mask" is a digital image file, which may be altered by the operator of the tool so that it compensates for the deformations introduced by processing steps (i.e. resist deposition, baking, handling, development, and etching). While flexibility is one of the appealing features of maskless lithography, cost reduction is another one, since expensive masks are not needed anymore in such a technology.

The micromirror array works like a mask to write patterns directly onto substrates. In comparison with other maskless lithography technologies, the micromirror based lithography technology possesses superior features. It is characterized by sufficient throughput for highly customized patterns, fine lithographic quality, efficiency in cost and time [8], and so on. Nevertheless, the micromirror based lithography is feasible if, and only if, each system developer could set up an excellent optic unit [9], and an accurate DMD control unit [10].

We combine this digital lithography masking method with each of the micro-fabrication technology to fabricate high-density micro-well arrays with improved spot uniformity, and micro-pillar array with resolution near of single micromirror size (one pixel size). The precision on combine method is shown by repeatability rate of the fabrication result.

DESIGN OF DIGITAL LITHOGRAPHY SYSTEM

We modified Digital Light Processing (DLP) commercial projector for maskless lithography system. It consists of robust projection and control system of digital micromirror device (DMD) array. The light source was replaced by high intensity UV light from INNO-Cure 100, which transfers through liquid light-guide to put inside the DLP projector. The proposed system than mount and operate on 3-axes stage, which take part on adjusting the focal position and flexibility on reaching larger workpiece area. With this system, it is convenient to actualize the process of exposal and optional to replace the digital mask. These can provide possibility for maskless lithography being further improved.

1. MOTION STAGE

In order to focus the beam from projection part on the substrates, the whole part was mounted on a motion stage and moved in position relative to the workpiece. A projector holder was design to adjust perpendicularity to the workbench. As shown in fig.1, by varying the position of the part to the samples the correct focal position was determined. The work pieces were mounted on a workbench over two axis motion stage and scanned on a plane perpendicular to the UV-beam. This will add flexibility in reaching larger area of scanning the workpiece.

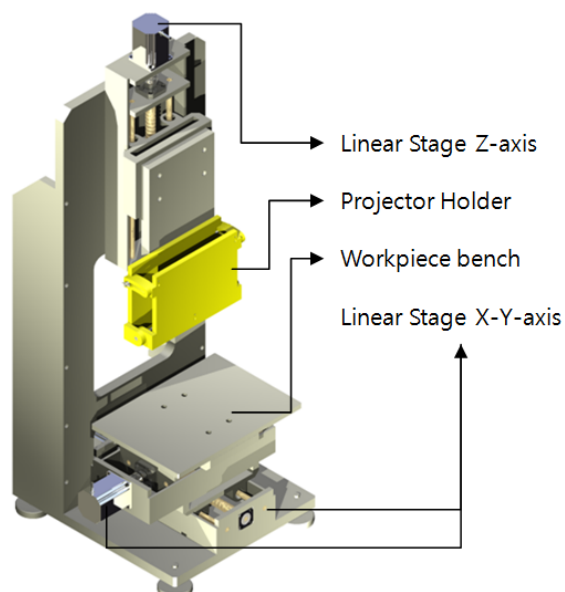


FIGURE 1. Mounting stage of DL system with projector holder for perpendicularity adjustment

2. PROJECTING SYSTEM

Commercial DLP projector consists of light source module, optical engine and control board of micromirror. Mercury lamp as light source module was removed and replaced by light-guide from the spot UV-source. The optical engine, as shown in fig. 2, consist of: 1) color wheel which function to generate color from white light source, 2) a collimator as the light tunnel, to aligned the light beams into direction of reflecting prism position, 3) Reflecting prism to redirect light onto DMD position. It make efficient space usage for compact design of the projector, 4) Shaping lens to fit the light beam with DMD target area, 5) Digital Micromirror Device (DMD) which define picture to be projected, 6) Projection lens to magnify the projected picture.

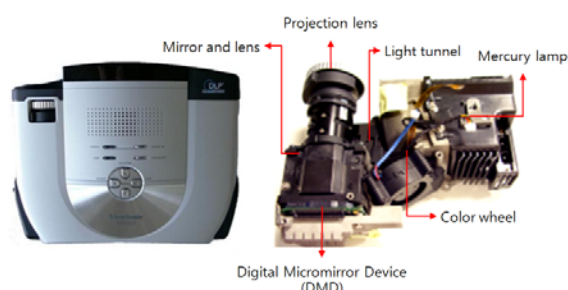


FIGURE 2. Optical engine of commercial projector.

We remove the color wheel because the UV-light no need to be generated into different colors. We also remove the projection lens in order to gain resolution as small as DMD pixel size, as shown in fig.3

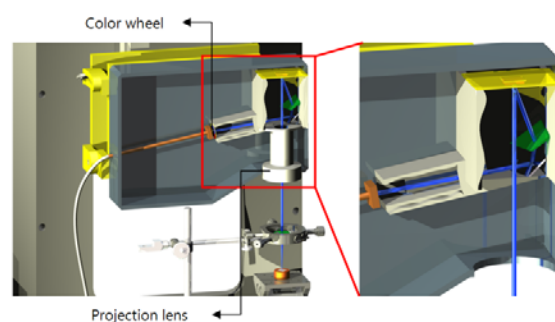


FIGURE 3. Modification of optical engine

The commercial projector has area of 1024x768 DMD array. The micromirror is $16 \times 16 \mu\text{m}^2$ and has separation of $1 \mu\text{m}$. Removing the objective length will allow to produce pattern same as micromirror size.

3. SIZE REDUCTION SCHEME

The shape of micromirror prevents to give edge free micro-pattern. To compensate it, we need to apply multi mirror for a micro-pattern. It mean the projected pattern will be bigger than $16 \mu\text{m}^2$ (theoretical resolution), as describe in fig.4.

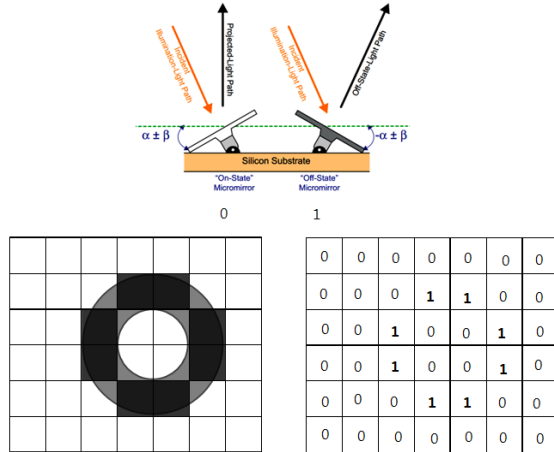


FIGURE 4. DMD size determined system resolution.

As we know, the relation between distance of pattern source to objective lens (S1) and the projected image to objective lens (S2) with focal point of the lens (f) is shown in eq.1:

$$\frac{1}{S_1} + \frac{1}{S_2} = \frac{1}{f} \quad (1)$$

Here, S1 is represent by the position of DMD on Z-axis and S2 by elevated workpiece.

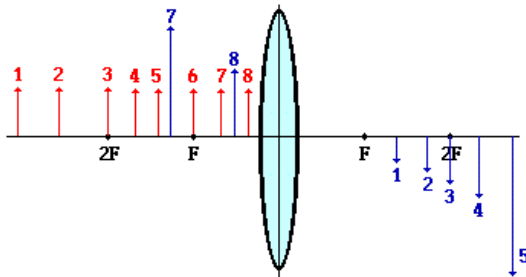


FIGURE 5. Relation between object position with related image..

From the ray diagram above, It showed that to real image with smaller size will appear if the object position is more than 2 times of focal point of the lens. If the magnification of the lens can be calculate from eq.2

$$M = \frac{S_2}{S_1} \quad (2)$$

Than by fixing the height of the workpiece surface (S2), we can get desired size of projected Image by adjusting the distance of DMD from objective lens using z-axis motion.

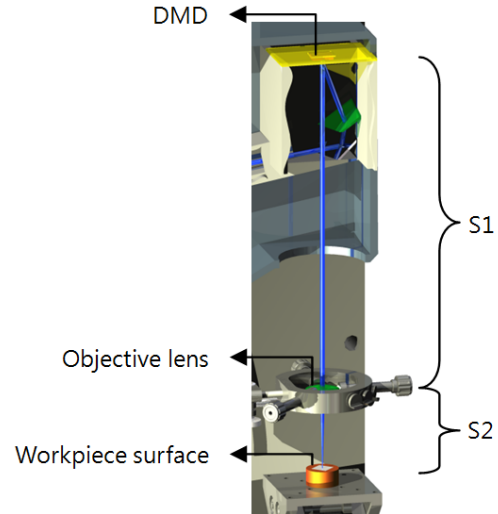


FIGURE 6. Adjustment mechanism to get real image on workpiece surface

BIOMACHINING OF MICRO-FEATURE ARRAY

Biomachining process was performed by placing photoresist masked workpiece in sterile jars filled with bacteria and culture medium. Microorganism we used is bacteria of *A. ferrooxidans* and SU8 2100 from microchem. The jars are then incubated in particular temperature for certain machining time.

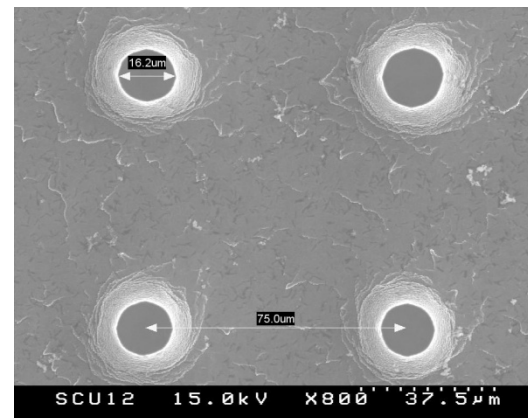


FIGURE 7. SU8 mask for micro-pillar-array

To avoid contamination and provide oxygen, the jars were closed but not tightly. The jars were incubated at 35°C without shaking for various machining times. After biomachining process, the workpieces were then removed, rinsed and dried. To analyze the final result, the scanning electron microscope (SEM) photograph was used.

The circular array or pillar was developed on Cu using biomachining process. The SEM photograph shows the size and shape of photoresist mask (Fig.7) and following biomachined micro-pillar array (Fig.8). The diameter of top section of pillar is about 16 μm and the distance between pillars was 75 μm .

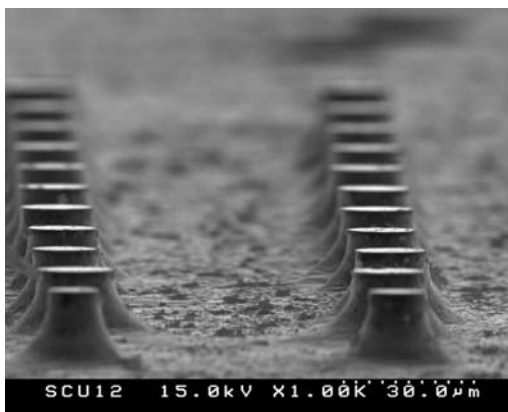


FIGURE 8. Micro-pillar-array by biomachining.

STATUS AND OUTLOOK

A DL system prototype is being constructed. The system consists of five main parts as shown in fig.9

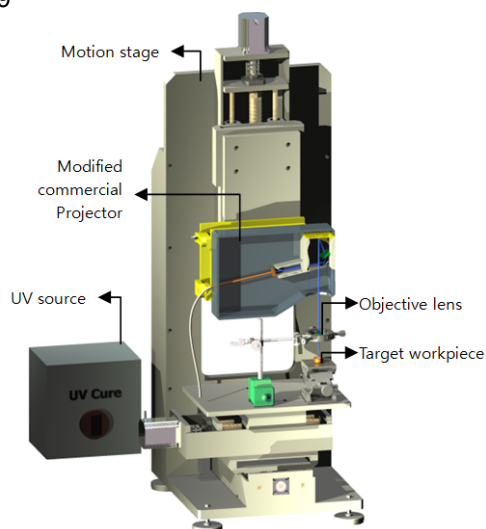


FIGURE 9. Solid model of DL system prototype

KEYWORDS

Uniform micro feature, Precise micro array, Digital lithography

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MACHINE TOOL GENOME PROJECT CASE STUDY

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INTRODUCTION

Milling process models may be implemented to enable pre-process parameter selection for optimized performance. To enable accurate process performance prediction using these models, the tool-holder-spindle-machine structural dynamics must be known [1]. The required tool point frequency response function (FRF) can be obtained by modal testing. However, for the large number of tool-holder combinations in typical production facilities, the measurements can be prohibitively time-consuming and costly.

In the Machine Tool Genome Project (MTGP)¹, Receptance Coupling Substructure Analysis (RCSA) is applied as an alternative to modal testing [2-4]. In the RCSA approach, the tool-holder-spindle-machine assembly is considered as three separate components: the tool, holder, and spindle-machine and the individual FRFs of these components are coupled analytically. The archived measurement of the spindle-machine FRF (or receptance) is coupled to the free-free boundary condition receptances of the tool and the holder derived from beam models. In the MTGP paradigm, the tool, holder, and spindle-machine are considered as “genes”. RCSA provides the “mapping” step to predict the “body characteristic” (assembly FRF). This paper presents a case study to demonstrate the utility of the MTGP to pre-process milling parameter selection. FRF and stability predictions are compared to experiments.

SETUP DESCRIPTION

Three nominally identical Haas TM-1 CNC machining centers were tested; see Fig. 1. The spindle-machine FRF was measured for all three using a simple geometry “standard holder”; see Fig. 2. The portion of the holder beyond the flange was removed in simulation to isolate the

spindle-machine dynamics and this result was archived. A Schunk Sino-R tool holder and M.A. Ford TuffCut GP two flute 12.7 mm x 25.4 mm (flute length) x 76.2 mm (overall length) endmill were modeled and coupled to the spindle-machine receptances. The tool overhang length was 50.9 mm. The connection stiffness between the tool and holder was identified in a separate setup using a 12.7 mm diameter carbide blank. This value is generic to all 12.7 mm diameter tools in this holder and was also archived.



FIGURE 1. Haas TM-1 CNC machining centers.



FIGURE 2. Standard holder used to measure spindle-machine FRF (CAT-40 interface).

¹ This research was, in part, funded by the U.S. Government. The views and conclusions contained in this document are those of the authors and should not be interpreted as representing the official policies, either expressed or implied, of the U.S. Government.

FRF PREDICTION AND MEASUREMENTS

Right machine

For the right TM-1 machine in Fig. 1, the spindle-machine receptances were coupled to the tool-holder model and compared to a measurement of the tool point FRF on this machine. The x-direction result is provided in Fig. 3 (prediction: blue, measurement: red).

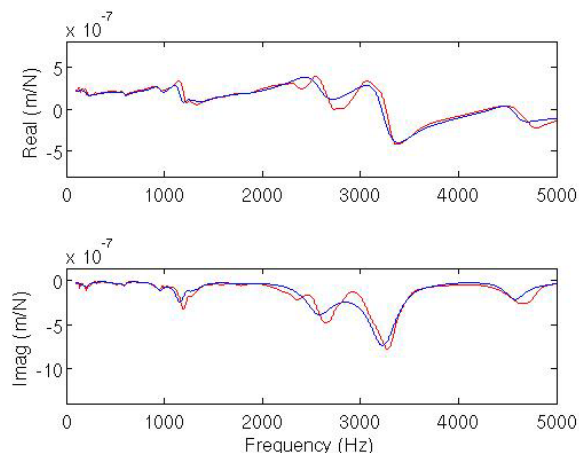


FIGURE 3. Right TM-1 comparison (right spindle-machine receptances).

Middle machine

The spindle-machine receptances for the middle TM-1 were coupled to the tool-holder model and compared to a measurement of the tool point FRF on the same machine. See Fig. 4.

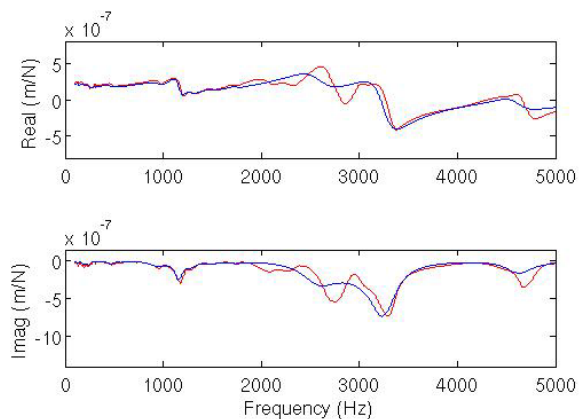


FIGURE 4. Middle TM-1 comparison (middle spindle-machine receptances).

Left machine

The spindle-machine receptances for the left TM-1 were coupled to the tool-holder model and

compared to a measurement of the tool point FRF on the same machine. See Fig. 5.

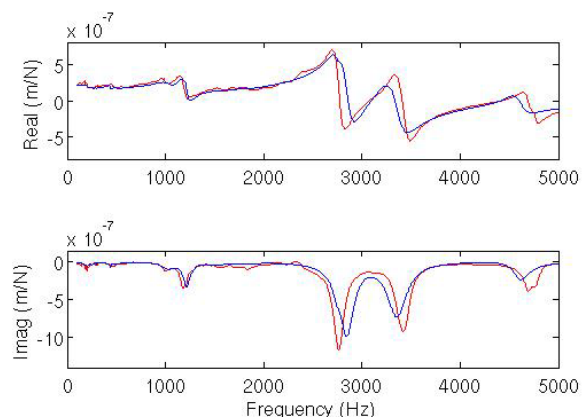


FIGURE 5. Left TM-1 comparison (left spindle-machine receptances).

To extend the analysis, predictions were next completed for the middle and left machines, but using the spindle-machine receptances for the right TM-1.

Middle machine-right machine receptances

The spindle-machine receptances for the right TM-1 were coupled to the tool-holder model and compared to a measurement of the tool point FRF on the middle machine. See Fig. 6. It is observed that the agreement between prediction and measurement is comparable to Fig. 4. This suggests that the spindle dynamics between the right and middle machines are similar.

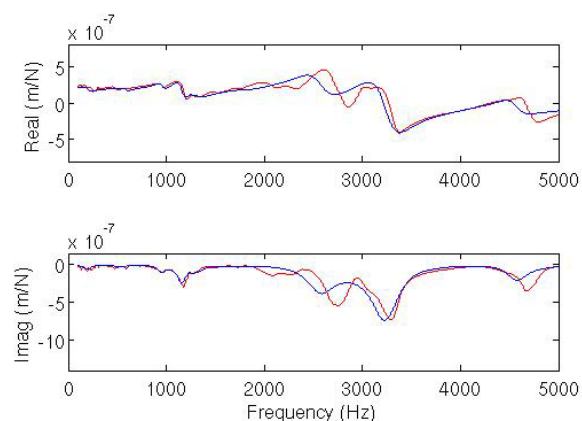


FIGURE 6. Middle TM-1 comparison (right spindle-machine receptances).

Left machine-right machine receptances

The spindle-machine receptances for the right TM-1 were next coupled to the tool-holder model and compared to a measurement of the tool point FRF on the left machine. See Fig. 7. The decreased agreement between prediction and measurement relative to Fig. 5 suggests that the spindle dynamics between the left and middle machines differ.

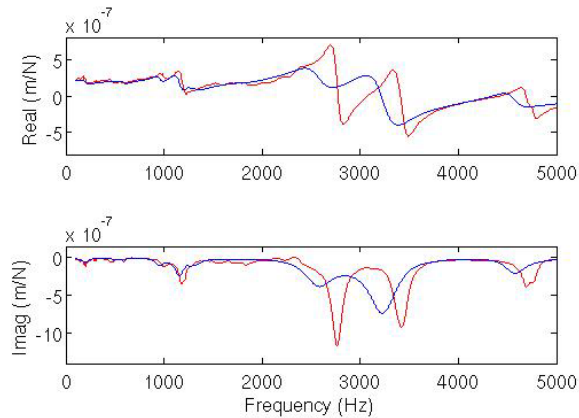


FIGURE 7. Left TM-1 comparison (right spindle-machine receptances).

STABILITY PREDICTION AND TESTS

Right machine

For the right TM-1 machine, the predicted and measured tool point FRFs in the x and y-directions were used to generate the stability lobe diagram for a 50% radial immersion up-milling cut in 6061-T6 aluminum. Figure 8 shows the limiting axial depth of cut, b_{lim} , for the predicted (blue) and measured (red) FRFs.

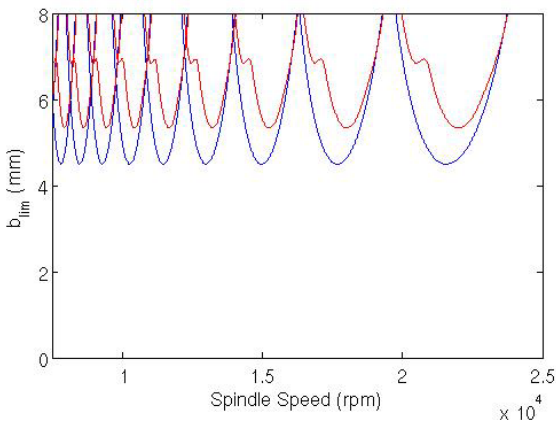


FIGURE 8. Left TM-1 stability comparison (right spindle-machine receptances).

Middle machine-right receptances

Next, the predicted tool point FRFs in the x and y-directions produced using the right machine spindle-machine receptances were used to generate the stability lobe diagram. Figure 9 shows b_{lim} for the predicted and measured (tool-holder in middle TM-1) FRFs. It is again observed that the “best speeds” are accurately identified and the minimum stability limit is conservative for the predicted result.

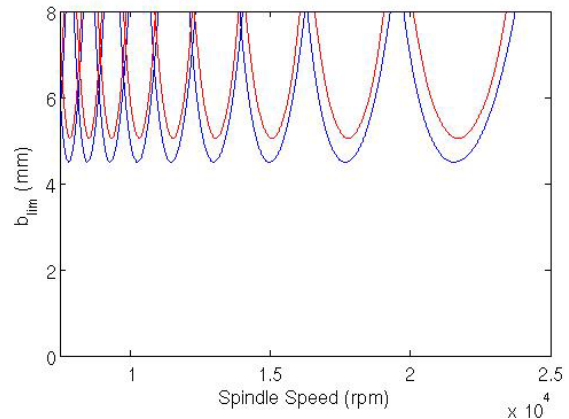


FIGURE 9. Middle TM-1 stability comparison (right spindle-machine receptances).

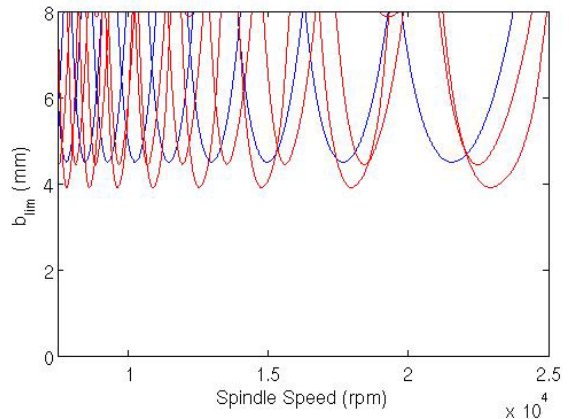


FIGURE 10. Left TM-1 stability comparison (right spindle-machine receptances).

Left machine-right receptances

Finally, the predicted tool point FRFs in the x and y-directions produced using the right machine spindle-machine receptances were used to generate the stability lobe diagram. Figure 10 shows b_{lim} for the predicted and measured (tool-holder in left TM-1) FRFs. The disagreement is a direct result of the FRF mismatch in Fig. 7.

Cutting tests

The middle TM-1 stability limit was predicted using the right machine spindle receptances. This boundary was then evaluated using cutting tests on the middle machine. Figure 11 shows the stability limit and test results: stable (o), unstable (x), and marginal ($\square$).

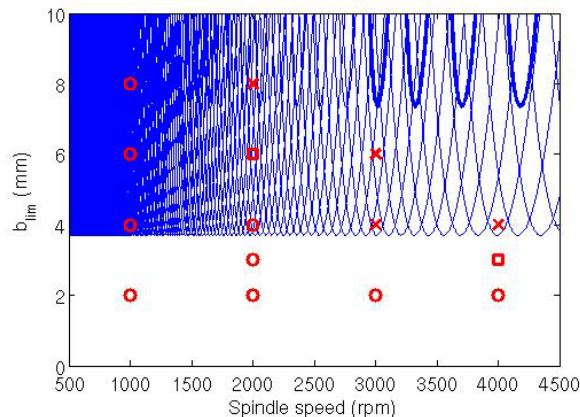


FIGURE 11. Middle TM-1 stability results (right spindle-machine receptances).

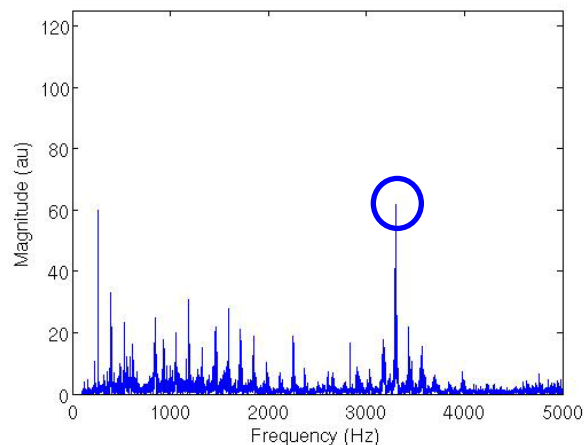


FIGURE 12. Spectral content of sound signal for unstable cut at {4000 rpm, 4 mm}. The chatter frequency near 3400 Hz is identified.

First, it is observed that considerable process damping is present at the 1000 rpm and 2000 rpm spindle speeds. The analytical stability model used here does not include this effect, so the trend of increasing stability with reduced cutting speed is not present in the prediction. At 4000 rpm, however, the experimental results confirm the stability prediction. The spectral content of the sound signal for the {4000 rpm, 4

mm axial depth cut} is provided in Fig. 12. The chatter frequency at approximately 3400 Hz corresponds to the most flexible vibration mode near 3200 Hz in Fig. 6. Conversely, the stable cut at {4000 rpm, 2 mm} does not exhibit content at this frequency; see Fig. 13.

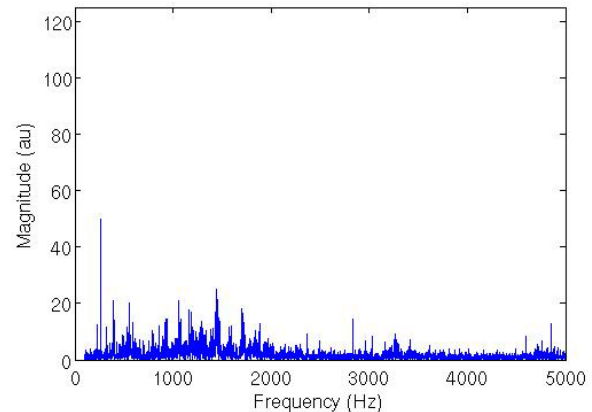


FIGURE 13. Spectral content of sound signal for stable cut at {4000 rpm, 2 mm}.

CONCLUSIONS

The Machine Tool Genome Project was introduced and experiments were used to validate the performance of the Receptance Coupling Substructure Analysis approach to select pre-process milling parameters.

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ENHANCEMENT OF POWER EFFICIENCY OF SOLID OXIDE FUEL CELLS BY ALIGNING MICROSTRUCTURE PATHS OF ION/ELECTRON/GAS IN ANODE

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1. INTRODUCTION

A fuel cell, which is an electric power generation device that converts chemical energy to electric energy directly, has been considered as an attractive device in recent years. A solid oxide fuel cell (SOFC) is one of the most promising energy conversion devices because of its higher power efficiency and fuel utilization ratio than other fuel cells. An SOFC consists of three layers: a cathode, an electrolyte and an anode. In the cathode, oxygen reacts with electrons from an external circuit to oxide ions. *FIGURE 1* shows the structure of the anode. The anode consists of three phases: yttria-stabilized zirconia (YSZ), Ni and pores. In the anode, a chemical reaction occurs at the point where these three phases come into contact (three-phase boundary: TPB), and oxide ions, electrons and gases pass through YSZ, Ni and pores, respectively [1].

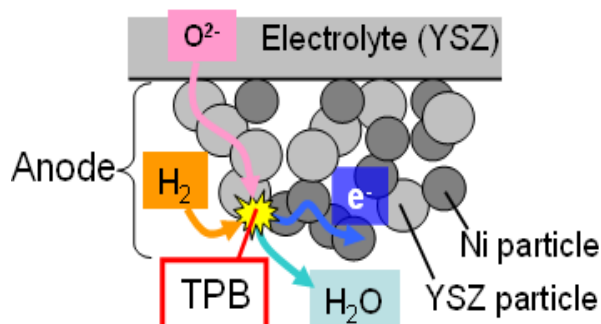


FIGURE 1. Structure of anode of SOFC

Our purpose is to align Ni and YSZ in the thickness direction in the anode to enhance the power efficiency of SOFCs. In a conventional anode, the three phases are uniformly distributed and the paths of oxide ions, electrons and gases have high resistances [2]. When Ni

and YSZ phases are aligned, the resistances of electrons, oxide ions and gases are reduced.

In this study, we propose a novel method to align the three paths. When a magnetic field is applied in the thickness direction, Ni particles magnetically polarize and attract each other. As a result, Ni particles are aligned in the thickness direction, as shown in *FIGURE 2*.

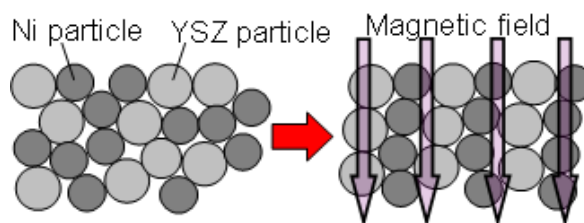


FIGURE 2. Alignment of Ni by magnetic field

2. EXPERIMENTAL

2.1. Preparation of Anode

Ni (average particle diameter: 1 μm) and YSZ (average particle diameter: 1.1 μm) particles were mixed at a 1:1 volume ratio to obtain a mixed powder. Terpeneol and methylcellulose were mixed at a 100:3 weight ratio to obtain a binder. The mixed powder and the binder were mixed at a 11:3 weight ratio. The mixture slurry was screen printed on YSZ electrolyte (thickness: 500 μm) with a thickness of 50 μm . A magnetic field was applied to the sample using two magnets located at the top and bottom of the sample. The intensity of the applied magnetic field was 360 mT. The sample was heated to 300 $^{\circ}\text{C}$ at a rate of 10 $^{\circ}\text{C}/\text{min}$ and maintained at 300 $^{\circ}\text{C}$ for 1 h with the magnets to evaporate the binder. After the magnets were removed, the sample was heated to 1400 $^{\circ}\text{C}$ at a rate of 10 $^{\circ}\text{C}/\text{min}$ and maintained at 1400 $^{\circ}\text{C}$ for 3 hours to sinter Ni and YSZ. Subsequently,

the cathode (lanthanum strontium manganate: LSM) was applied to the other side. The sample was heated to 1200 °C at a rate of 10 °C/min for 6 h to sinter LSM. Finally, NiO, which was formed during sintering, was reduced to Ni by annealing in H₂ atmosphere.

2.2 Reconstruction of Microstructure of Anode

The method of reconstructing the microstructure of the anode was reported by Iwai *et al.* [3] and Shikazono *et al.* [4]. Using this method, we reconstructed the microstructure of the anode.

2.2.1. Observation by Focused Ion Beam-Scanning Electron Microscopy (FIB-SEM)

The pore in the anode was filled with an epoxy resin under vacuum. After the epoxy was cured, the sample was roughly polished with sandpaper. Then the cross section was polished with an Ar ion beam cross-section polisher (CP). Carbon was coated on the polished cross section at thicknesses of 2-3 μm to improve the electrical conductivity.

The sample was observed by FIB-SEM (SII-NT SMF1000). The principle of FIB-SEM is shown in FIGURE 3. The acceleration voltage of SEM was 1.0 kV during the observation. By scanning FIB as shown in FIGURE 3, three hundred SEM images of the cross sections of the anode were observed. The pitch between the observed cross sections was 50 nm.

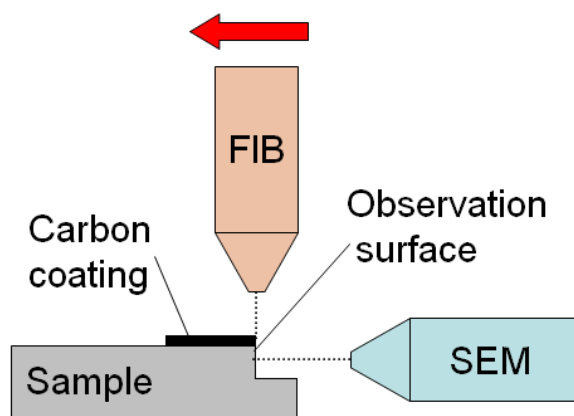


FIGURE 3, Scheme of FIB-SEM

2.2.2. Microstructure Reconstruction

To alter the misalignment between SEM images caused by the drift of SEM, the three hundred SEM images observed were aligned, and the necessary areas in each SEM image were trimmed. FIGURE 4 shows an SEM image of the cross section of the anode after alignment

and trimming. In the images, the bright, middle and dark parts indicate Ni, YSZ and pores, respectively.

Subsequently, the 3 phases were completely separated by the brightness value. The brightness values of Ni, YSZ and pores after separation were 0, 127 and 255, respectively. FIGURE 5 shows an image of cross section of the anode after the separation of the 3 phases.

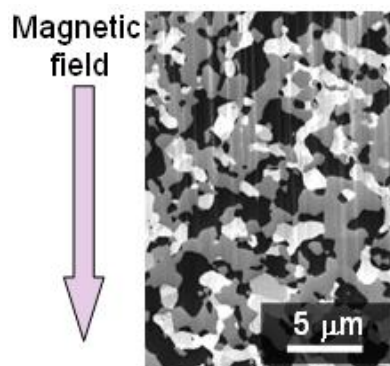


FIGURE 4, SEM image of cross section of anode after alignment and trimming

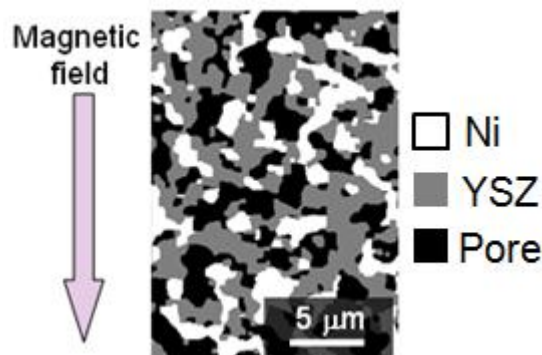


FIGURE 5, Image of cross section of anode after separation of 3 phases

Finally, the 3D microstructure of the anode was reconstructed. FIGURE 6 shows the reconstructed anode microstructure. The images of the cross sections of the anode after separation were laminated in the z direction. In FIGURE 6, the green and blue parts indicate Ni and YSZ, respectively. The size of the voxel is 75 nm on one side. The thickness and magnetic field direction is x-axis direction in FIGURE 6.

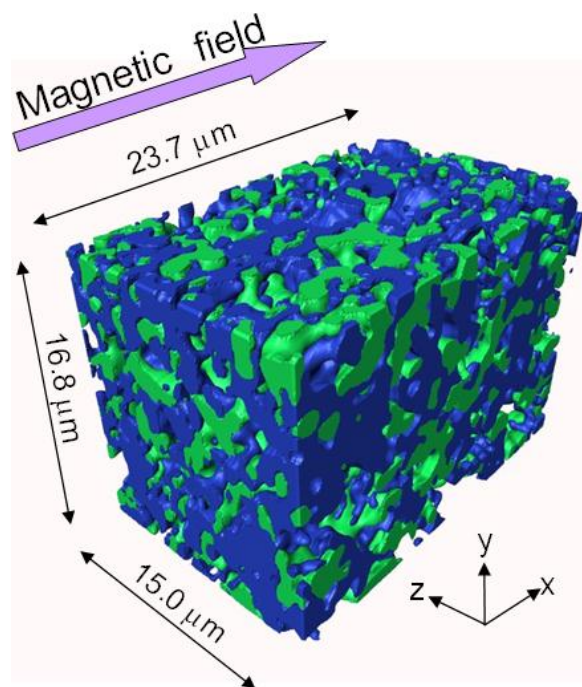


FIGURE 6, Reconstructed microstructure of anode (blue: YSZ, green: Ni)

2.3. Calculation of Structure Parameters

To estimate the electrode performance, the TPB and the active TPB densities were calculated by the centroid method proposed by Shikazono *et al.* [4].

The midpoints of the edges shared by Ni, YSZ and pore voxels located next to each other were identified. Then the centroids of the triangles defined by the neighboring three midpoints mentioned above were connected by a line. The length of this line was defined as the TPB length. The TPB length per unit volume was calculated as the TPB density to compare samples of different volumes.

The active TPB is defined as the TPB which connects to the electrolyte ($x=0 \mu\text{m}$), current collector ($x=23.7 \mu\text{m}$) and bulk gas phase ($x=23.7 \mu\text{m}$) through YSZ, Ni and pore phases, respectively.

2.4. Evaluation of Power Density

The electric power density was measured using an SOFC evaluation device (BEL SOFC, BEL Japan, Inc.). The electric potential between the anode and the reference electrode was measured as a function of the current density.

3. RESULTS and DISCUSSION

3.1. TPB and active TPB densities

TABLE 1 shows the TPB and active TPB densities of an anode with a magnetic field and a conventional anode [4].

The TPB density in the anode with a magnetic field was almost the same as that of the conventional anode. The active TPB density in the anode with a magnetic field was 1.2 times higher than that in the conventional anode.

The ratio of active to total TPBs in the anode with a magnetic field was 1.3 times higher than that in the conventional anode.

This is because Ni particles were magnetically polarized by the magnetic field and attracted each other, and the connectivity of Ni particles increased. Moreover, the connectivity of Ni particles from TPB to the current collector increased.

3.2. Examination of Power Density

The relationship between the current density and the voltage between the anode and the electrolyte is shown in FIGURE 7.

The relationship between the current density and the power density is shown in FIGURE 8. The approximated curves shown in FIGURE 8 by a dashed line were calculated by the least-squares method, and the maximum power density was calculated using the approximated curves. The maximum power density with a magnetic field of 360 mT was 1.6 times higher than that without a magnetic field.

This is because the ratio of the active TPB density increased in the presence of a magnetic field.

TABLE 1. TPB and active TPB densities

	TPB density [$\mu\text{m}/\mu\text{m}^3$]	Active TPB density [$\mu\text{m}/\mu\text{m}^3$]	Ratio of active TPBs
Anode with magnetic field	2.39	1.90	79.5 %
Conventional anode [4]	2.56	1.54	60.2 %

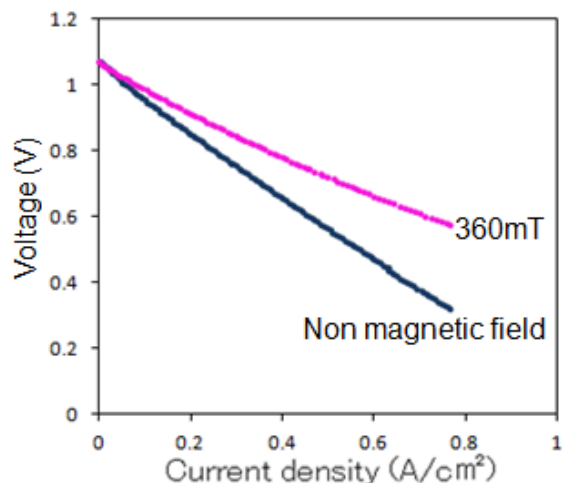


FIGURE 7, Relationship between current density and voltage between anode and reference electrode

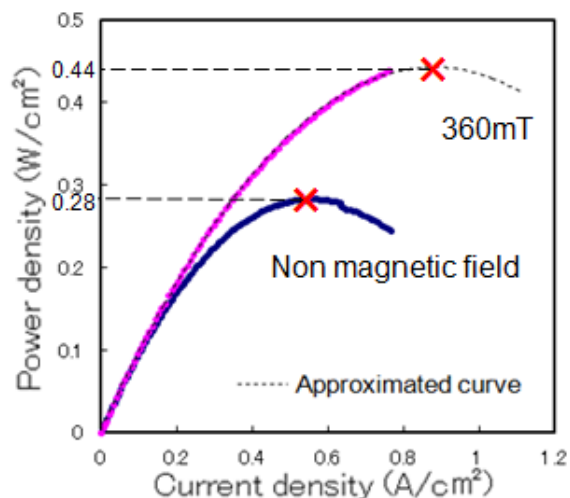


FIGURE 8, Relationship between current density and power density

4. CONCLUSIONS

In our research, the microstructure of Ni-YSZ anode was controlled by applying a magnetic field. The following results were obtained.

The active TPB density in an anode with a magnetic field was 1.2 times higher than that in a conventional anode. The ratio of active to total TPBs in the anode with a magnetic field was 1.3 times higher than that in the conventional anode.

The maximum power density with a magnetic field of 360 mT was 1.6 times higher than that without a magnetic field. The electrode performance was improved by applying a magnetic field.

ACKNOWLEDGMENT

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Adhesion Control of Coating Film by Texturing

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INTRODUCTION

The surface of most engineering products, such as cars and machine parts, is coated as coating is an important process which improves visual and tactile quality and provides preservative effects of the products. However, there are no reports on fundamental high quality coating conditions such as coating injection quantity or coating times for products with various types of surface textures.

The purpose of this study is thus to detect optimal coating conditions to realize high quality coating, by verifying the relation between coating characteristics and surface textures using the coating robot developed by the authors.

In the previous report, we reported on the results of measuring coated metal sheet surface roughness with a scanning white light interferometer and coating film adhesion by cross cut tests [1]. In this work, we carried out coating of different types of textured metal sheets and evaluated coating adhesion by cross cut test, cylindrical mandrel bend test, and pull off test.

DEVELOPED COATING ROBOT

Figure1 shows an overview of the developed coating robot. This robot is composed of two electrically powered sliders along the X and Y axes and a spray gun installed on the X axis. The speed of the sliders and the ON/OFF switch of the spray gun can be controlled by computer. The maximum coating distance on the X axis is 350 mm, while that on the Y axis is 400mm. The coating speed range is between 10 to 200 mm/s.

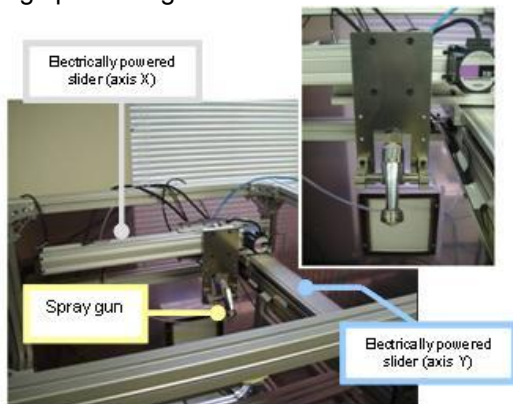


Figure 1. Overview of developed coating robot

COATING OF NON-TEXTURED METAL SHEET AND EVALUATION OF VISUAL AND TACTILE QUALITY

Experiment method

We coated metal sheets (cold rolled steel) using the developed coating robot, and evaluated the tactile quality of the coating by measuring the coated sheet surface roughness using a scanning white light interferometer. To evaluate the visual quality, we measured the coated surface glossiness with a gloss meter. Three types of coatings were performed on the metal sheets; under, top, and clear coating. Each coating was jetted for one minute and then the injection quantity was calculated from the weight change of the coatings. Air pressure was set at 0.3MPa and coating speed at 50mm/s in these experiments.

Verification experiments on difference in surface roughness and glossiness according to coating times

(1) Difference in metal sheet surface roughness according to number of under and top coatings

Under and top coatings were carried out several times on 50 x 50mm metal sheets, and the difference in the coated surface roughness was measured with a scanning white light interferometer. Under coating was carried out N_u times and top coating for N_t times on 20 sheets where $0 \leq N_u \leq 4$ and $1 \leq N_t \leq 4$. All the sheets were also given two clear coatings. Table 1 shows the surface roughness (Sq) of the sheets before and after coating. In Table 1, the surface roughness (Sq) before coating is indicated in blue, and that after is indicated in black. Based on Table 1, Figure 2 shows metal sheets with two under, top, and clear coatings.

As a result, the surface roughness after coating was less than $0.31\mu\text{m}$ and smoother than that before coating. The surface roughness of metal sheets that had been coated four times was rougher than others, and the surface of metal sheets without under coating was rougher than those that had under coating. The smoothest coated surface was obtained when $N_u=2$ and $N_t=2$. The surface roughness in this case was $0.31\mu\text{m}$.

Table 1. Surface roughness (Sq) before and after coating

Root mean square (RMS) roughness (Sq) Unit : μm		Number of top coatings			
		1	2	3	4
Number of under coatings	0	1.35	1.36	1.33	1.34
		0.56	0.58	0.71	0.67
	1	1.32	1.38	1.36	1.35
		0.39	0.36	0.45	0.55
	2	1.37	1.34	1.32	1.30
		0.32	0.31	0.39	0.75
	3	1.40	1.35	1.34	1.31
		0.40	0.43	0.41	0.87
	4	1.32	1.34	1.33	1.38
		0.67	0.78	0.69	0.89

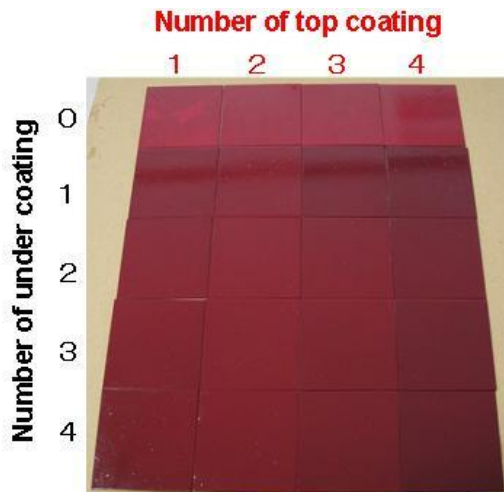


Figure 2. Metals sheets with two under, top, and clear coatings

(2) Evaluation of glossiness of metal sheets according to number of clear coatings

We next verified the difference in the coated surface roughness and glossiness according to the number of clear coatings. Under coating was carried out 0 to 3 times, and top coating 1 to 4 times on 50 x 50mm metal sheets. After two clear coatings, glossiness was measured using a gloss meter, and the average of three sheets was taken as the glossiness. Table 2 shows the glossiness after coating. Based on Table 2, Figure 3 shows that $N_u=2$ and $N_t=2$ for sheets with clear coating. As a result, the glossiness value of all sheets with clear coating was high, from 8 to 13%, except for sheets without under coating. Glossiness was found to increase with the number of clear coatings, and was above 88% in all the metal sheets which had three clear coatings.

Table 2. Glossiness after coating

Number of Under coatings	Number of top coatings	Number of clear coatings			
		0	1	2	3
0	1	37.1	50.4	80.3	90.5
	2	34.8	53.1	84.2	89.5
	3	35.6	49.9	88.4	88.1
	4	36.3	54.1	85.3	93.9
1	1	12.8	45.3	75.6	91.4
	2	11.8	48.8	77.1	90.3
	3	10.5	52.4	69.4	89.9
	4	9.7	48.5	75.4	86.3
2	1	11.3	44.9	69.8	88.1
	2	11.7	45.2	70.9	91.4
	3	10.3	48.1	73.7	93.2
	4	8.9	43.2	72.9	90.3
3	1	9.7	44.3	73.8	88.3
	2	9.4	39.1	75.9	87.0
	3	8.8	48.2	78.2	90.2
	4	9.0	41.9	69.3	91.5

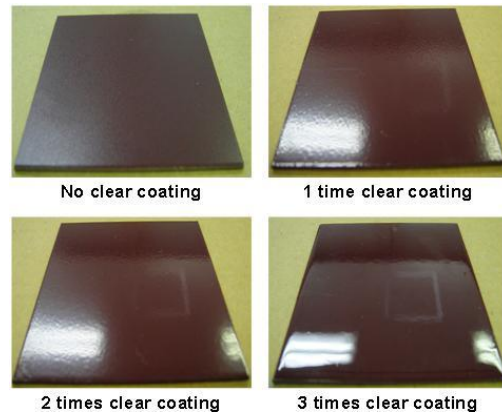


Figure 3. $N_u=2$, $N_t=2$ after clear coating

COATING OF TEXTURED METAL SHEETS AND EVALUATION OF COATING ADHESION

Experiment method

We performed coating on textured metal sheets, and verified the difference in coating adhesion according to texture pattern and coating direction. We then evaluated coating adhesion by cylindrical mandrel bend test [2] and cross cut test [3] qualitatively, and by pull off test [4] quantitatively. The cylindrical mandrel bend test involves bending coated metal sheets and evaluating the coating. With the cross cut test, two series of parallel cuts cross angled to each other are cut on a coated metal sheet using a cutter to obtain a pattern of 25 squares, then tape is applied to and peeled off from the cuts to evaluate the adhesion of the coating. With the pull off test, the coating is pulled off from the metal sheet to measure the force required to break the coating.

Evaluation by cylindrical mandrel bend test

First three patterns were created on 50 x 100mm metal sheet surfaces as shown in Figure 4 by texturing in three ways, ①one direction, ②two directions (90° cross), ③two directions (45° cross), at six texturing levels using 80, 150, 220, 500, 800, and 1200 grit sandpapers. Next, under and top coatings were carried out twice each, as these provided the best coating results in the previous section, and the adhesion of the coating to the substrate was measured by the bend test. We used three kinds of mandrels with different diameters for this test; 6mm, 4mm, and 2mm. Generally, the shorter the diameter of the mandrel, the easier does the coating peel off. As an example, Figure 5 shows the results of the bend test for a 90° two-direction textured metal sheet using 80 grit sandpaper. Figure 5 shows pictures of the sheet after the bend test and the magnification of the bend of the sheet. As a result, it was found that the coating did not peel off in all types of metal sheets, indicating that the coating had sufficient flex resistance.

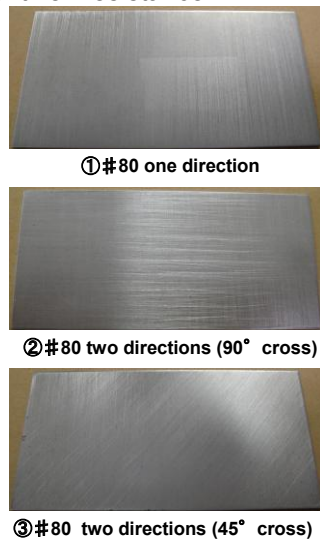


Figure 4. Metal sheet textured with 80 grit sandpaper

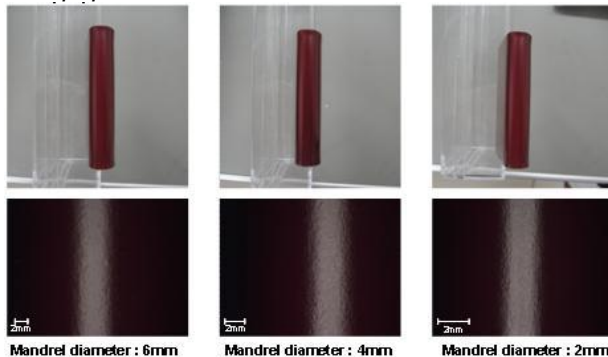


Figure 5. Results of bend test in two directions (90° cross) ② of metal sheet textured by 80 grit sandpaper

Evaluation by cross cut test and pull off test

Like the previous section, three texturing patterns were created on 50 x 50mm metal sheet surface; ①one direction, ②two directions (90° cross), ③two directions (45° cross) at six texturing levels using different sandpaper. After texturing three sheets, under and top coatings were carried out twice on each sheet, and coating adhesion was evaluated by cross cut test and pull off test. Each sheet was then graded between 0 to 5 according to the ISO standard. Figure 6 shows the results of evaluation by the cross cut test and pull off test, and Figure 7 shows some examples of the metal sheet surfaces after the cross cut test. As a result, it was found that coating adhesion was poorer with sandpaper with smaller grit. Compared with non-textured sheet, sheets textured using sandpaper of 80, 150, and 220 grit had poorer adhesion. Coating peeled off in most of these sheets, indicating insufficient adhesion.

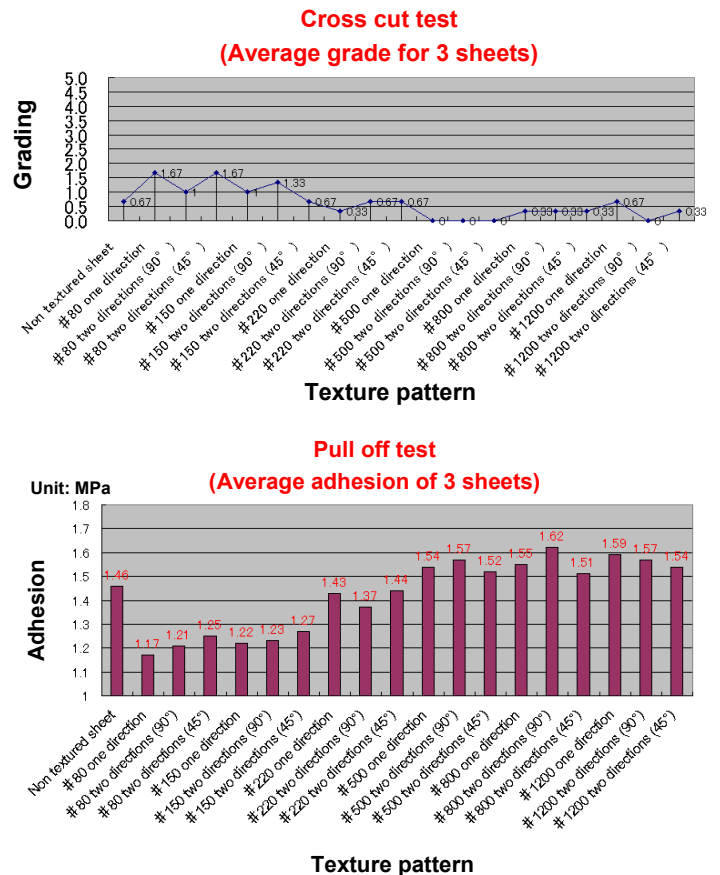


Figure 6. Evaluation results by cross cut test and pull off test

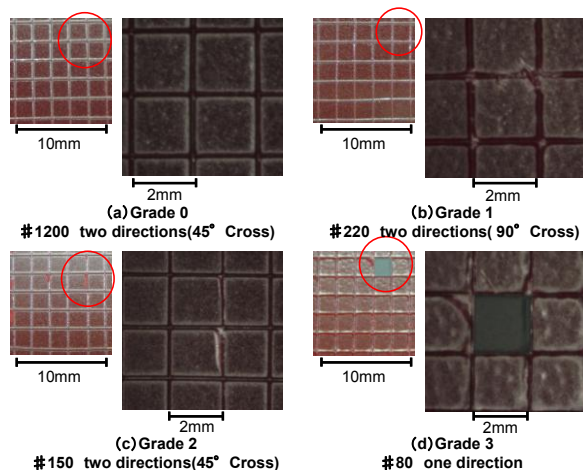


Figure 7. Metal sheet surfaces after cross cut test

Difference in adhesion according to coating direction

First, three texturing patterns were created on 50 x 50mm metal sheet surfaces; ①one direction, ②two directions (90° cross), ③two directions (45° cross) with 80 grit sandpaper and can opener.

Next under, top, and clear coatings were carried out twice for each sheet; (1) coated in the direction opposite to texturing, (2) coated in same direction as texturing, and evaluated coating adhesion by pull off test. Figure 8 shows the metal sheets textured using a can opener and coating direction. In the evaluation of the pull off test, adhesion values of five sheets were averaged. Figure 9 shows the result of the pull off test. For ③two directions (45° cross), there were no adhesion differences between (1) coated in the direction opposite to texturing and (2) coated in same direction as texturing. However, good adhesion was obtained for ①one direction and (2) coated in same direction as texturing. Furthermore, it was found that as the texturing depth of the metal sheet became deeper, the difference in the adhesion values became larger.

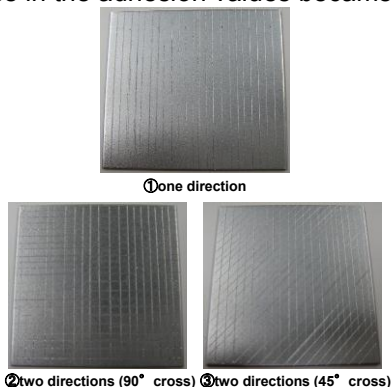


Figure 8. Textured metal sheets with can opener and coating direction

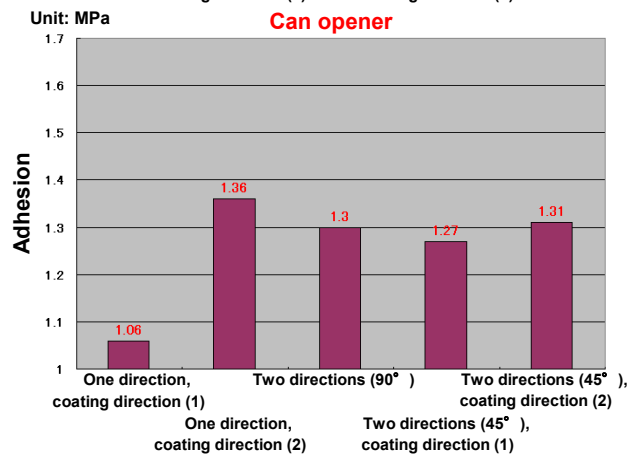
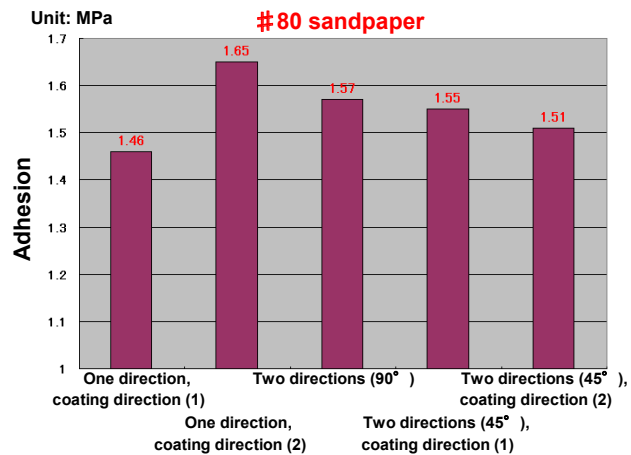


Figure 9. Results of pull off test

CONCLUSION AND FUTURE WORK

In this study, various types of textured metal sheets were coated using the coating robot developed by the authors, and coating adhesion was evaluated by the cylindrical mandrel bend test, cross cut test, and pull off test. As a result, it was found that coating along the same direction as texturing provided good adhesion. In future works, we plan to continue studies on coating textured metal sheets to verify high quality product coating conditions.

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Ultraprecision Engineering Approach in the Quantum Nano-joint Thomson Elements

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Introduction

The thermoelectric conversion that uses thermoelectric phenomena of the solid-state component can be converted directly into the electrical energy that uses huge unused thermal energy easily, and high expectations are drawn as a zero emission power generation technology that doesn't accompany the noise, the vibration, and the exhaust thing at all. This paper is first review the theoretical background of practical use research, the creation of the quantum nano-joint Thomson element of the size of nano with the possibility that the electric power that is continuous day and night and equals sunlight power generation can be invented is aim. Moreover, the metallic involving fullerene is referred.

Quantum Nano-joint Thomson Elements [1]

Figure 1 is a cooling system using two Nano-joint Thomson elements. As for the Ni filamentation, when the DC voltage is impressed to the Thomson element that consists of the Ni filamentation, the cooling part and the Ni filamentation become generation of heat parts with Ni filamentation and NiO (p-type semiconductor) filamentation. It is an atomic boundary side of the Ni filamentation and the thin metallic wire. As for the Cu filamentation, when the DC voltage is impressed to the Thomson

element that consists of the Cu filamentation by 10mV, the cooling part is caused in the generation of heat part and the Cu filamentation with Cu filamentation and ZnO (n-type semiconductor) filamentation. The surface temperature that consists of two Thomson elements and the polymer membrane or the glass substrate sides in the state of the normal temperature of 20°C is 8°C, and it operates as a cooling system. Moreover, when the cooling part of the nano-joint Thomson element that

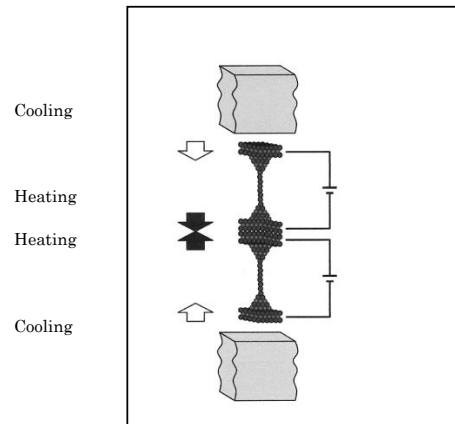


Fig.1 Quantum nano-joint Thomson elements

consists of p-type semiconductor in the cooling part of the nano-joint Thomson element that consists of n-type semiconductor comes in contact, the thermoelectric conversion device that works as a generation of heat system can be achieved.

Two personal computers shown in Figure 2 is made a parallel connection, and a high-speed simulation module has

been achieved.

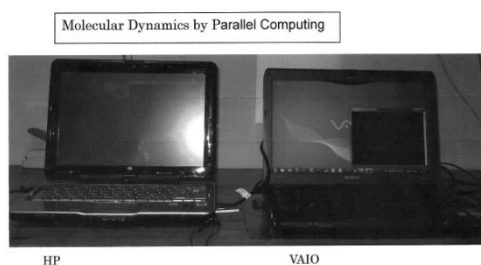


Fig.2 Parallel computing

Scanning probe lithography [2, 3]

The majority of the transcriptional pattern is exposed to one of the approaches that improve the productivity of the SPM lithography by the high throughput lithography of an optical lithography etc., and the compound lithography to which it draws in the part where a high resolution at the nano level is necessary by SPM is shown in Figure 3.

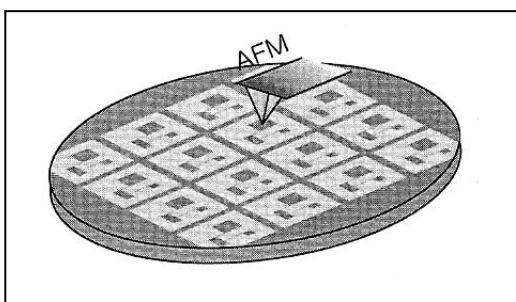


Fig.3 Scanning probe lithography.

1. The SiO_2 film is formed on the Si substrate.
2. Resist is spread on the SiO_2 film on the Si substrate.
3. V ditch pattern of width $1\mu\text{m}$ and $0.5\mu\text{m}$ in depth is exposed by the scanning probe lithography.
4. The SiO_2 film on the Si substrate is

made L letterform mask, and the Si substrate is processed to L letterform V ditch by anisotropy Etching.

The nano thin film is piled up in L letterform V ditch on the Si substrate with the arc plasma assistance device.

- ① L letterform both wing part where the Thomson element is composed is separated with the mask, and the division metallic nano thin film is piled up individually.

The nano thin film part piled up is covered with the oxide film.

- ② The semiconductor nano thin film is piled up in the center part of beer hall L letterform.

- ③ Electroconductive of the joint of the structures piled up in L letterform V ditch by a scanning probe (nano thin film + semiconductor nano thin film + nano thin film) is checked.

The quantum nano-joint Thomson element is completed in the above-mentioned process.

Ultraprecision Positioning [4]

Positioning high resolution that makes the best use of the characteristic of an enough stroke and the piezo unit obtained with the displacement expansion mechanism in the stage used for the micro processing of SPM is demanded. Therefore, it is thought that the development of the slight movement mechanism that combines the displacement expansion mechanism and

the displacement reduction mechanism and achieves the length stroke and the high resolution positioning by one element is useful. Moreover, a continuous positioning drive is important in the processing. Figure 4 is a structural outline of the slight movement device, measures, and shows the system. The slight movement displacement is taken out by using the beam with the lever mechanism with the elastic hinge. PZT1 and PZT2 are set up respectively on base 1 and base 2. The slight movement control is done by the warp detection. The displacement resolution makes the number of steps 300 nm and each number of five steps by this device, and can position a continuous step where the precision position decision of the slight movement of the reduction and the stroke of the slight movement of the expansion are united by being able to set up to 1 nm, and using the technique of a coordinated slight movement.

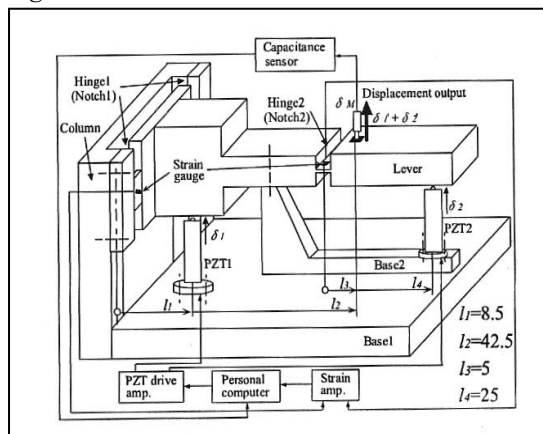


Figure 4 is a structural outline of the slight movement device, measures, and shown system.

Semimetal atoms involving fullerene Thomson element

Figure 5 is semimetal atoms involving fullerene Thomson element while developing. The method by which a control of an atomic level of the thin film and an extremely minute individual metallic involving fullerene can be made by high reproducibility includes the method of irradiating the plasma ion and the molecular operation method. This semimetal atom involving fullerene is made by the method of irradiating the plasma ion. Semimetal atom involving fullerene 100 was made by the method by which a control of an atomic level of the fullerene and an extremely minute individual metallic (semiconductor) involving fullerene were able to be made by high reproducibility. Semimetal atom involving fullerene 100 takes the structure that hydrogen atom 102 unites in, that is, one of two brachium conjunctivums of semiconductor Ge in the fullerene husk atom 101 the basket. One electron is insufficient in this connecting structure, and like a positive electron completely becomes and insufficient Ge atom 101 becomes p-type semiconductor atom. One electron is insufficient in this connecting structure, and like a positive electron completely becomes and insufficient Ge atom 101 becomes p-type semiconductor atom. Hydrogen atoms 104, 105, 106 of three per three brachium conjunctivum unite with Ge atom 103 in

the fullerene husk. Because a lot of one electron with a negative charge has it, the electron increases too much the number of free electrons and becomes n-type semiconductor atom. The state of the pn junction has been achieved in this semimetal atom involving fullerene 100. In the element involving fullerene, when the DC voltage is added from the outside to this semimetal atom involving fullerene 100, Thomson effect at an atomic level is caused, and generation of heat and endothermic are caused at both ends of the fullerene. In the element involving fullerene, when the DC voltage is added from the outside to this semimetal atom involving fullerene 100, Thomson effect at an atomic level is caused, and generation of heat and endothermic are caused at both ends of the fullerene. When 5mV was impressed to element involving fullerene 100 by the simulation by first principle molecular dynamics as 107 in the DC voltage, the current that flowed to the nano joint Thomson element was 2nA. The cooling part and the generation of heat part of temperatures fluctuate 10°C are caused at both ends of semimetal atom involving fullerene 10. It can be widely applied that the type semiconductor atom of n that does phosphorus and boron and did the ion driving to the Si atom that is the metallic element as this semimetal atom involving fullerene and the oxide semiconductor atom of the atom of the

semimetal in the fullerene husk of p-type semiconductor atom in the heat oxidation furnace furthermore.

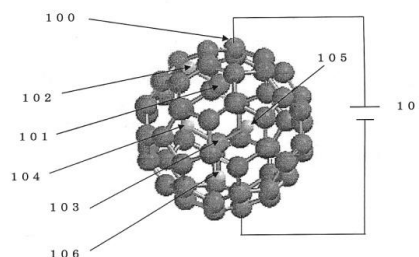


Fig.5 Semimetal atoms involving fullerene Thomson element

Conclusions

This paper proposes the cooling system using two quantum nano-joint Thomson elements. The semimetal atoms involving fullerene Thomson element will be expected as a thermoelectric conversion material in the future.

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Investigation of the Effects of High-Deformation, High-Precision Manufacturing Processes on Material Properties

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INTRODUCTION

High-energy-density experiments at the Lawrence Livermore National Laboratory's National Ignition Facility (NIF) and the Laboratory for Laser Energetics's Omega laser require targets containing polished tantalum foils with a two-dimensional sine-wave pattern imprinted on one side (Figure 1). The peak-to-valley amplitudes and wavelengths of the sine waves are on the order of 5 μm and 50 μm - 100 μm , respectively. Surface finish requirements are on the order of 20 nm R_A , and the thickness of the 3 - 5 mm diameter foils are 30 μm - 50 μm . Manufacturing these high-precision features is challenging because the tolerances are not achievable with conventional machining, and tantalum is a material that cannot be single-point diamond turned. Furthermore, physics requirements for these targets specify use of a fine-grain, or single-crystal, high-purity tantalum starting material that is not available in foil geometry; rolling the available material would distort the grain structure and render the material unacceptable. Therefore, these samples are manufactured by mechanically polishing discs to the appropriate thickness and then coining (embossing) the pattern onto the surface using a nitrided-steel die with the sine-wave pattern diamond turned onto it. Both polishing and coining impart mechanical damage on the tantalum, and the mechanical properties and grain morphology of the material can be altered. Here, we report on our efforts to quantify changes in material properties as a result of these manufacturing processes. Because these foils are manufactured for surface-sensitive experiments that probe mechanical properties at high pressures and high-strain rates, it is important to understand and quantify how fabrication processes affect the material, particularly if the effects are prominent at the surface. These parts are subject to tight dimensional tolerances, requiring careful metrology; the additional materials specifications imposed by the experiments require that "traditional" metrology techniques

are supplemented by materials science characterization tools.

RESULTS AND DISCUSSION

Mechanically polishing metals to thicknesses on the order of tens of microns is challenging, as this dimension approaches the grain size and thus results in an extremely fragile foil. Furthermore, a foil containing only a few grains through the thickness may exhibit different mechanical properties than that of a bulk material because dislocations encounter fewer impediments as they move through the sample. In mechanically polished polycrystalline tantalum foils with an average grain size of $16 \pm 6 \mu\text{m}$, we observe a distinctive banding pattern when the thickness is reduced to approximately 50 μm . These deformation bands are observed with scanning electron microscopy on both sides of the foil and through the cross-section (Figure 2). When the sample is vacuum annealed at 980°C for 30 minutes the bands disappear. Using electron backscatter diffraction (EBSD), it was determined that these bands are small-angle ($< 5^\circ$) misorientations within the grains (Figure 3). We hypothesize that the thin geometry allows dislocations to run virtually unimpeded through the thickness of the foil. Although these deformation bands look similar to crystallographic twins, the small misorientation angle is inconsistent with twinning; in body-centered cubic materials such as Ta, the misorientation angle for twins would be 60° . While these small misorientations are acceptable for polycrystalline Ta targets, a change in strength resulting from the deformation (work hardening) may be undesirable. We are planning nanoindentation experiments to measure potential strength (hardness) changes caused by the deformation banding that results from mechanical polishing to very thin dimensions.

After the tantalum is polished to the correct thickness and surface finish, a sine-wave pattern is imposed on the surface by coining. This

process can also alter mechanical properties because it introduces severe plastic deformation, particularly at the patterned surface. Nanoindentation revealed that the hardness of the coined region was approximately 25% higher than that of the surrounding uncoined area. It is not yet known how far this manufacturing “damage” extends into the thickness of the tantalum foil, but cross-sectional indentation experiments to quantify these effects are planned.

Single-crystal (100) and (111) oriented tantalum samples were also coined with a 4 μm peak-to-valley sine-wave pattern. For these parts, maintaining the correct crystallographic orientation at the surface of the material is essential, as the parts are used in experiments that measure properties as a function of crystallographic orientation. Using EBSD, we determined that coining caused regions within the single crystal to rotate as much as 13° (Figure 4). The crystallographic damage imparted by the coining extends through the thickness of the parts, although it is most severe on the slopes of the sine-wave ripples. In the (111) oriented sample, rotations on the flat side of the part are more severe than what is seen on the (100) oriented sample. While the orientation may play a role in the extent of the damage, the older, slightly deformed die used to coin the (111) sample probably imparted additional damage on the back side of the sample, causing

the rotations. The crystallographic damage caused by the coining process is undesirable for these parts. However, the EBSD quantification of the rotations allows the experimenters to build this damage into their models. These parts are examples of how materials science characterization and more traditional metrology tools must work together to ensure that high-precision parts meet all experimental specifications.

CONCLUSIONS

This work demonstrates the value of connecting materials science characterization with precision manufacturing. Specifications for high-precision features are often accompanied by stringent requirements for materials properties, such as grain size and hardness. For some high-precision parts, characterizing the materials properties can be of equal importance to metrology of part dimensions. During manufacturing, one must be careful to assure that processing does not introduce undesirable characteristics in the material. Alternatively, deformation imparted by manufacturing processes could be utilized to produce unique material changes with beneficial implications for the part.

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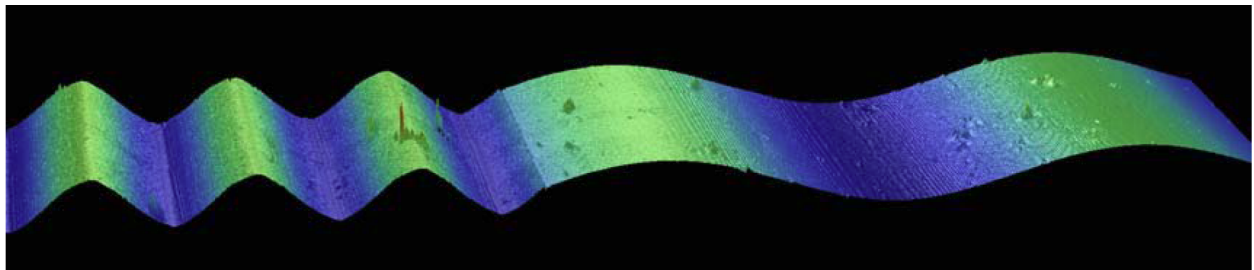
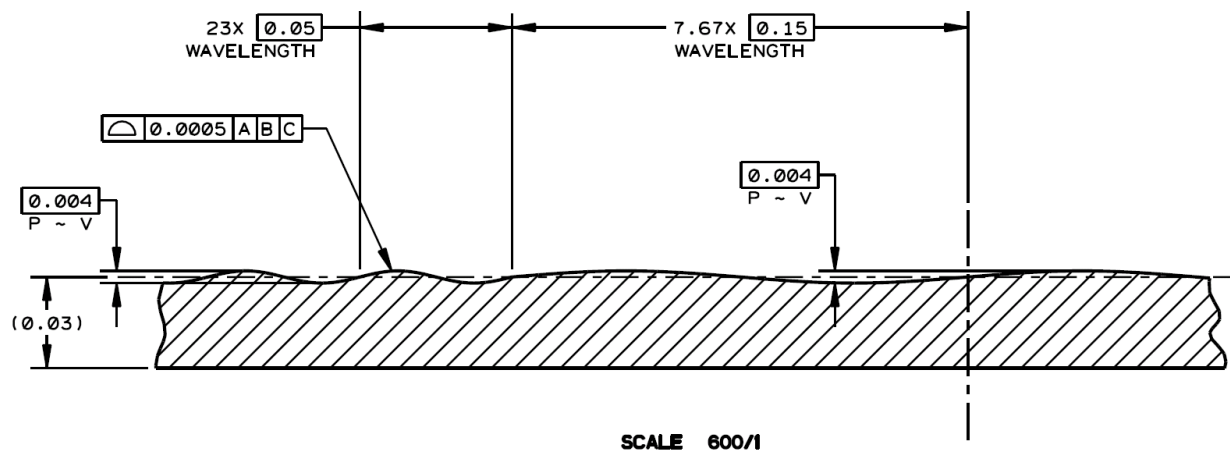


Figure 1: Top: Mechanical drawing of showing the sine-wave patterns required on a tantalum foil; bottom: white-light interferometer data from a coined tantalum sample with two adjacent sine-wave patterns.

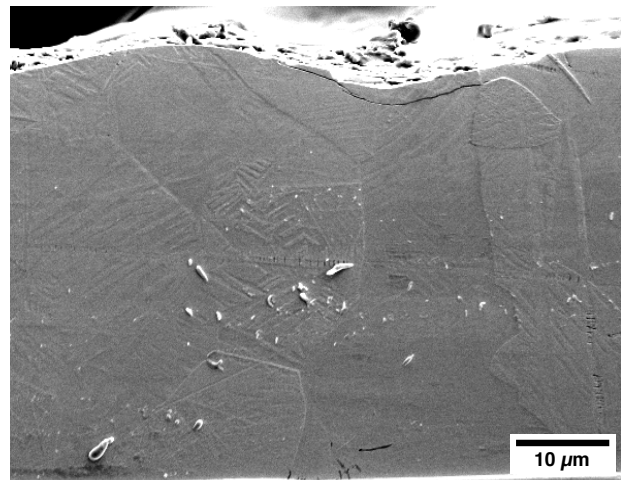
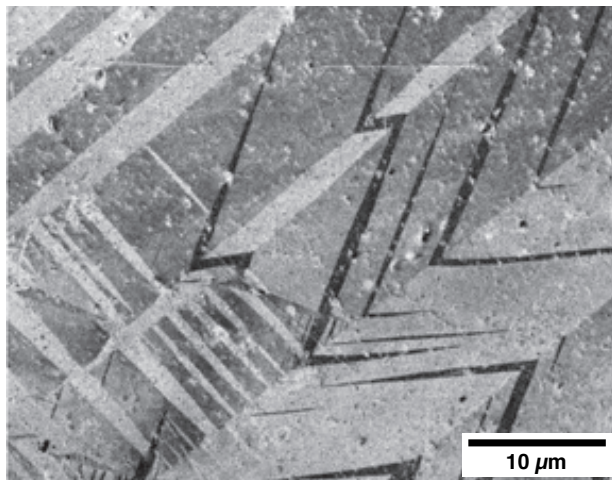


Figure 2: Scanning electron microscopy images of a polished (left) and coined (right) tantalum foil. The zig-zag features are deformation bands that form when the foil is polished to a thickness of less than 50 μm.

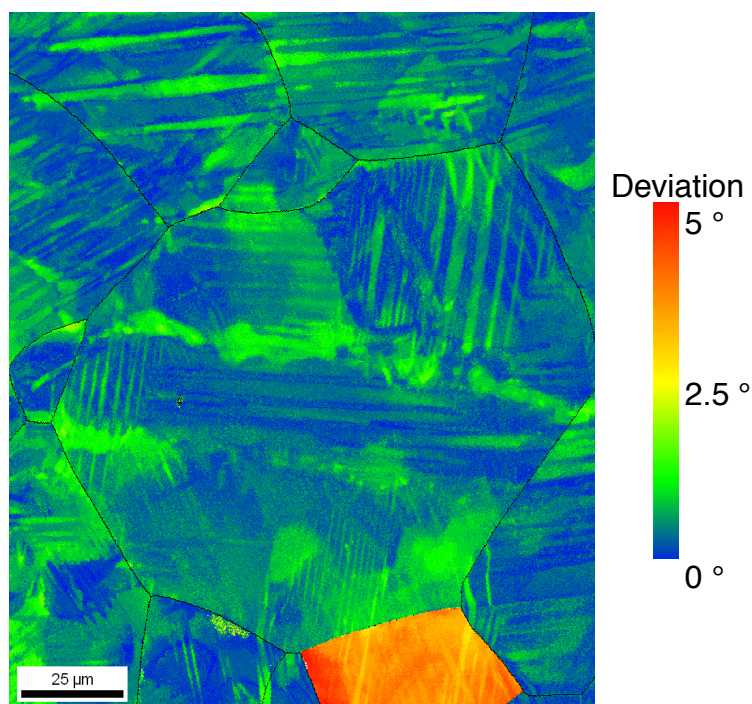


Figure 3: Electron back-scatter diffraction image of deformation bands in a 50 μm thick tantalum foil. The misorientation between the bands and the matrix is on the order of a few degrees, indicating that the bands are not twins.

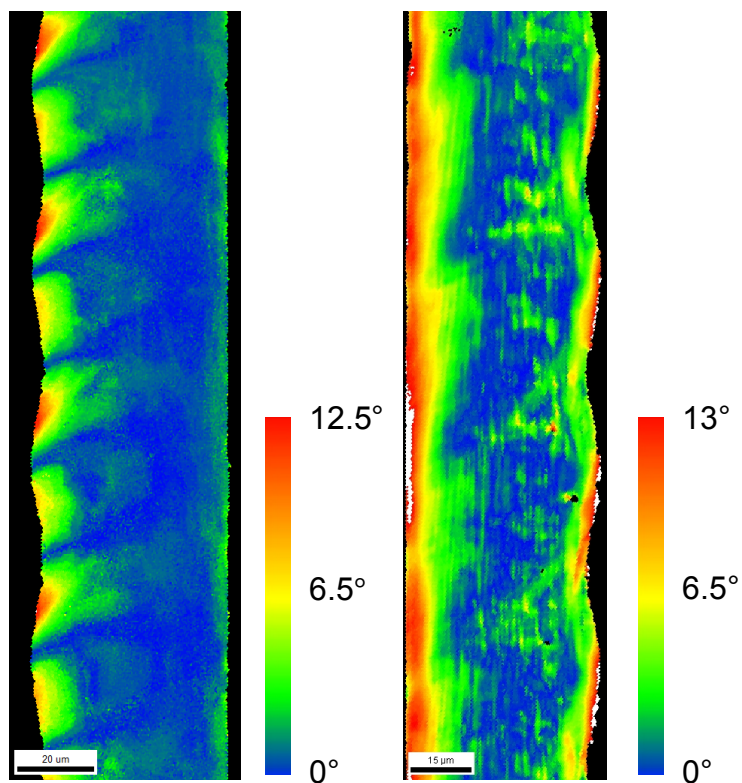


Figure 4: EBSD images of (100) (left) and (111) (right) coiled tantalum. Colors indicate the amount of misorientation from the matrix orientation. The ripples contain regions that are crystallographically misoriented by as much as 13° from the single-crystal matrix.

MICROMIXER OF MULTI-MATERIAL SLURRY WITH ROTATION-SPLIT-COMBINATION CYCLE FOR CERAMIC COMPONENTS

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1. INTRODUCTION

High-functional ceramic components, such as nozzles of ink-jet printers, electrodes of fuel cells, printed circuits, heat exchangers, and diesel particulate filters are produced from ceramic slurry, i.e., a mixture of ceramic particles and a solvent. To develop new mixture components from many types of ceramic materials, such as Al_2O_3 , ZrO_2 , SiC, SiN_4 , and TiN, a high efficiency mixing device is required. We developed a micromixer in which multiple slurries can be mixed quickly and continuously. Microscale mixing is more effective than macroscale mixing because the ratio of surface area to volume is larger in the microscale. Mixing can be defined as making the area of the boundary surface of different materials larger.

We designed a micromixer to perform a rotation-split-combination cycle. Figure 1 shows the schematic of our micromixer. First, multiple slurries are made to flow into a channel (1). Second, the slurries are rotated 90° by the oblique ridges fabricated on the channel wall (2). Third, the slurries are split at the center (3). Fourth, the split slurries are rotated 90° in each channel (4). Finally, the rotated slurries are combined (5). The number of layers of different slurries doubles through the cycle. Therefore, the number of layers is increased exponentially by repeating the cycle.

The amount of ceramic powder that naturally

diffused in slurry is very small. For example, a SiC particle in slurry whose size is about $1\ \mu\text{m}$ can diffuse only at a distance of several particles. Therefore, when SiC slurry is made to flow into the micromixer with a thickness of $20\ \mu\text{m}$, through 4 cycles, the thickness of one layer becomes $1.25\ \mu\text{m}$, which is smaller than the diffusion length, and a complete slurry mixing is achieved.

It has already been demonstrated that a Newtonian can be mixed with a micromixer [1]. However, multiple slurries have not been used. It is difficult for different material slurries to flow uniformly in a microchannel because of their viscosity characteristics. In this study, we mixed SiC and YSZ slurries with our micromixer.

2. EXPERIMENTAL

2.1. Diffusion of SiC Slurry and YSZ Slurry

We determined the number of SiC and YSZ particles that diffused naturally. We had SiC slurry and YSZ slurry get in touch each other on a plate. After drying the slurries, we observed the boundary of the two particles by scanning electron microscope (SEM).

2.2. Fluxion of SiC and YSZ Slurries in Microchannel without Pattern.

We observed the fluxion of the SiC and YSZ slurries in a microchannel without a pattern. SiC and YSZ slurries have different viscosity

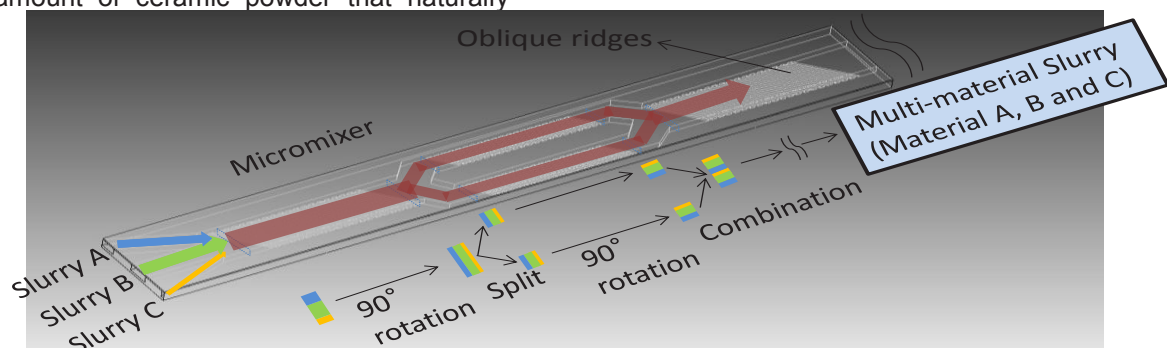


FIGURE 1. Schematic view of the micromixer.

characteristics. Therefore, we determined the densities of the SiC and YSZ slurries, and flow pressure so that the slurries can flow uniformly in the microchannel. The uniform flow in the microchannel is characterized by the same flow rate of the SiC and YSZ slurries.

2.3. Rotation of SiC and YSZ Slurries

We observed the fluxion of the SiC and YSZ slurries in a channel with oblique ridges to ensure the simultaneous rotation of the SiC and YSZ slurries by oblique ridges and measure the length of the channel required for 90° rotation. First, we made the SiC and YSZ slurries flow into the channel uniformly. Then, we observed the fluxion with a microscope.

2.4. Split and Combination of SiC and YSZ Slurries

We observed the fluxion of the SiC and YSZ slurries in a channel with split and combination parts to ensure that the two slurries split and combined appropriately. In this experiment, we used a channel with split and combination parts but without oblique ridges. We made the SiC and YSZ slurries flow into the channel uniformly in the same manner as that in experiment 2.3. Then, we observed the fluxion with a microscope.

2.5. Mixing of SiC Slurry and YSZ Slurry

We attempted to mix the SiC and YSZ slurries with the micromixer for the rotation-split-combination cycle. We determined the length of the channel with oblique ridges for 90° rotation from the results of experiment 2.4. In this experiment, we attempted to mix the SiC and YSZ slurries through two cycles. We made the SiC and YSZ slurries flow into the micromixer uniformly in the same manner as that in experiments 2.3 and 2.4. Then, we observed the fluxion with a microscope.

We also observed the distribution of the SiC and YSZ slurries that flowed from the micromixer to measure how well we mixed the SiC and YSZ slurries. We placed drops of slurry that flowed from the micromixer on a glass slide and observed the contact surface of the slurry and the glass slide with a microscope. We compared the distribution of the SiC and YSZ slurries on the contact surface between the no cycle, one cycle and two cycle cases.

3. RESULTS AND DISCUSSION

Figure 2 shows the SEM image of the boundary of SiC and YSZ particles obtained in experiment 2.1. The distance between the red and blue lines shows the distance at which the SiC and YSZ particles can naturally diffuse. This shows that the SiC and YSZ particles gradually diffuse naturally. This is due to the high viscosities of the slurries.

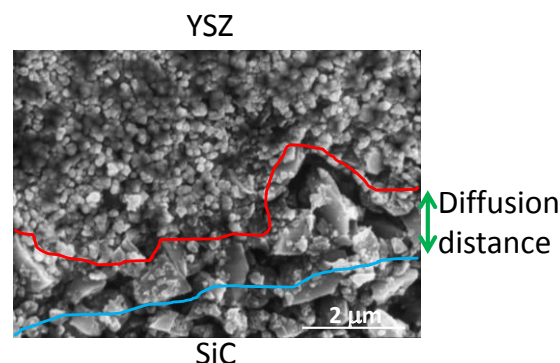


FIGURE 2. SEM image of boundary of SiC and YSZ particles.

Figure 3 shows the fluxion of SiC and YSZ slurries in the microchannel without a pattern observed in experiment 2.2. We made SiC and YSZ slurries flow into the microchannel at the same rate when the flow pressure was 0.16 MPa. However, the flow rate of YSZ slurry decreased and that of SiC slurry increased as they flowed at a pressure of 0.23 MPa. This was because the SiC and YSZ slurries had different viscosity characteristics. Generally, the viscosity coefficient of slurry depends on shear velocity. The relationship between viscosity coefficient and shear velocity differs between the SiC and YSZ slurries. The viscosity coefficients of the SiC and YSZ slurries change as flow pressure changes because shear velocity changes as flow pressure changes. When flow pressure is appropriate and their viscosity coefficients are equal, the SiC and YSZ slurries flow at the same rate. When flow pressure is not appropriate and their viscosity coefficients are different, the SiC and YSZ slurries flow at different rates.

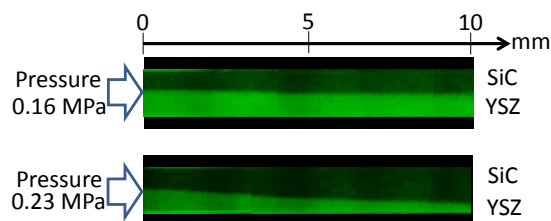


FIGURE 3. Fluxion of SiC and YSZ slurries in microchannel without pattern.

Figure 4 shows the channel with oblique ridges used in experiment 2.3 and the fluxion of SiC and YSZ slurries in the channel. It was observed that the SiC and YSZ slurries rotated simultaneously in the channel, and the length of the channel for 90° rotation was determined to be about 4.5 mm for the channel of 250 μm width and about 8 mm for the channel of 500 μm width.

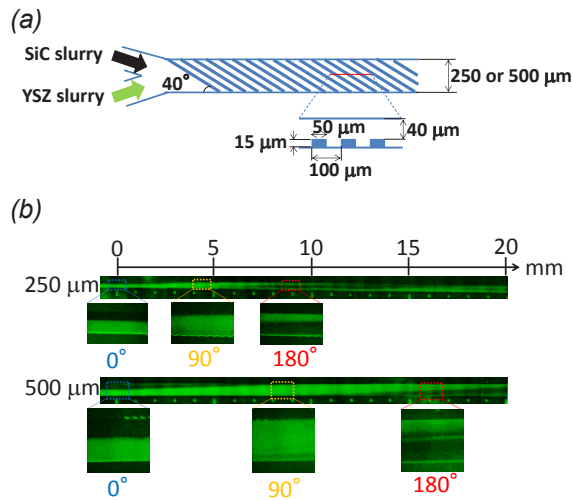


FIGURE 4.
(a) Schematic view of channel with oblique ridges used in experiment 2.3.
(b) Fluxion of SiC and YSZ slurries in channel with oblique ridges.

Figure 5 shows the fluxion of the two slurries in the channel with split and combination parts used in experiment 2.4. We observed that we can split and combine slurries appropriately using the shape of the channel. We showed that

the SiC and YSZ slurries little mixed by flowing in the channel. Because of the high viscosities of the slurries and the microscale channel, the flow remained as laminar flow throughout the channel containing split and combination parts and the two slurries little mixed through the channel.

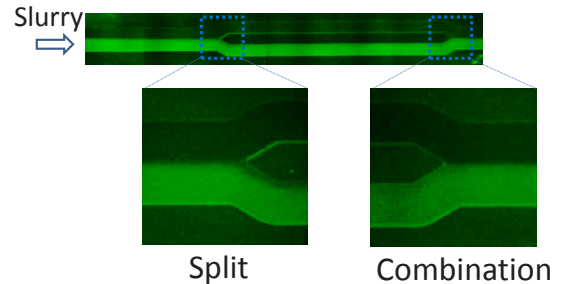


FIGURE 5. Fluxion of SiC and YSZ slurries in channel with split and combination parts.

Figure 6 shows a schematic of our micromixer used in experiment 2.5 and the fluxion of the SiC and YSZ slurries in the micromixer. We observed that the number of layers of the SiC and YSZ slurries doubled successfully through one cycle and increased further through two cycles.

Figure 7 (a) shows the image of the contact surface of the glass slide and the slurry drop that flowed from the micromixer. We observed that the SiC and YSZ slurries separated clearly in the no cycle case and that they appeared to be better in the two-cycle case than in the no-cycle and one-cycle cases. Figure 7 (b) shows the histogram of the image of the contact surface. The histogram of no-cycle case has two peaks

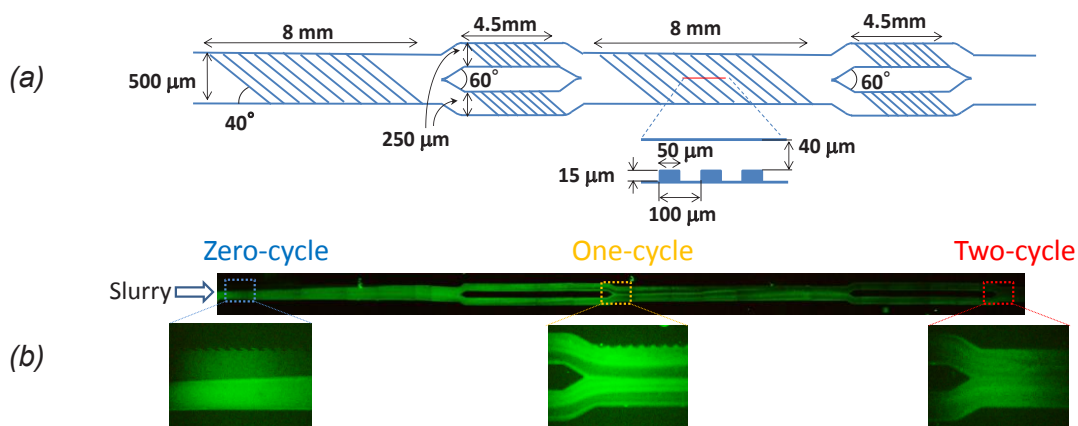


FIGURE 6.
(a) Schematic view of micromixer used in experiment 2.5.
(b) Fluxion of SiC slurry and YSZ slurry in the micromixer.

at about 20 and 60, indicating that the SiC and YSZ slurries separated clearly. The histograms of one-cycle and 2-cycle cases have a peak at about 45. The histogram of 2-cycle case has a sharper peak than that of 1-cycle case, indicating that the SiC and YSZ slurries mixed better in 2-cycle case than through 1-cycle case.

σ in Figure 7 (b) is the standard deviation of the histogram. σ decreases in every cycle, indicating that the SiC and YSZ slurries mixed through the cycle.

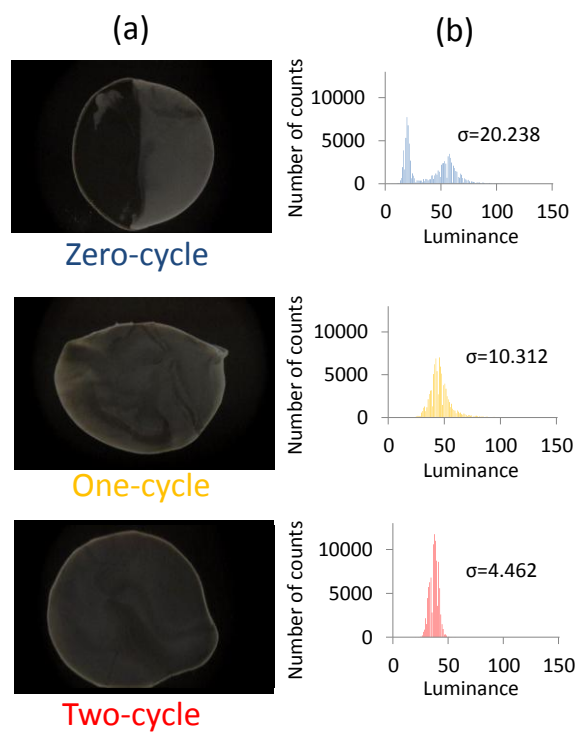


FIGURE 7.

(a) SEM image of contact surface of glass slide and slurry drop that flowed from micromixer.

(b) Histogram of luminance of image of slurry(σ is the standard deviation).

4. CONCLUSION

We demonstrated that there was an appropriate flow pressure for the SiC and YSZ slurries to flow into a microchannel at the same rate. This was because the viscosity coefficient of slurry depends on shear velocity and the relationship between viscosity coefficient and shear velocity is different between the SiC slurry and YSZ slurries.

We also demonstrated that the number of layers of the SiC and YSZ slurries in our micromixer

doubled successfully through one cycle and increased further through two cycles. We estimated how well our micromixer mixed the SiC and YSZ slurries using the histograms of the luminance of the slurry image and its standard deviation. The histogram of 2-cycle cases had a sharp peak at about 45 and the standard deviation decreased in every cycle. These results show that the mixing of the SiC and YSZ slurries was promoted by our micromixer.

We could mix the SiC and YSZ slurries using our micromixer this time. We now aspire to mix more types of slurry at once at any slurry ratio.

ACKNOWLEDGMENT

The photomasks for the microchannels used were fabricated using on EB lithography machine, F5112+VD01, Advantest Corporation, donated to the VLSI Design and Education Center (VDEC), the University of Tokyo.

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Visual Inspection of Free Form Glossy Surfaces using Phase Shifted Deflectometry

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Introduction

Phase shifted deflectometry is an optical metrology technique developed by Markus Knauer et. al. as a method to measure specular free-form surfaces[1]. The difficulty seen by many industries in objectively viewing these surfaces to find defects is being able to determine changes in gloss levels. With no real quantifiable method of gloss detection available, visual inspectors must be used to accomplish this task. This can lead to a wide range of quality in throughput even when stringent standard requirements are put in place. This effect can compound when production is done on a global scale, in multiple locations, and with visual inspectors receiving different training and understanding as to what level of defect quality is acceptable. With the advancement of phase shifted deflectometry methods, automated visual inspection can now be done to reduce these variances and increase the overall efficiency of material flow and standard of quality during production.

PRINCIPLE OF OPERATION

Phase shifted deflectometry acts on the principle that one is observing the surface of a test piece illuminated by a pattern and not measuring the actual light reflected off its surface. By viewing the actual test piece it is possible to evaluate the sample reliably even under conditions that the object is not in the same position each time. Additionally, the light that is reflected diffusely from the surface of the object is still able to be imaged when viewing the pattern on the object, rather than requiring the rays to be directed into the camera's sensors.

The phase shifted deflectometry system adapted for visual inspection consists of a white light LED array that generates continuous illumination patterns. A four-axis computer with a numerically controlled positioning stage moves the part under test into optimum viewing angles. Also included are various cameras whose

optical resolutions are matched to the limits of the sought-after defect sizes and whose pixel resolutions satisfy the Nyquist-Shannon sampling theorem. Finally, a high performance computing cluster is added and scaled accordingly to allow for any evaluation algorithms to be performed at the speed of image acquisition such that evaluation times are not increased by the complex algorithms used to evaluate the parts under test.

The system's LED array produces a series of sinusoidal fringe patterns that are viewed by a camera as a reflection from the object under test (FIGURE 1). The series of patterns produced by the illumination element are shifted in phase and recorded by the system.

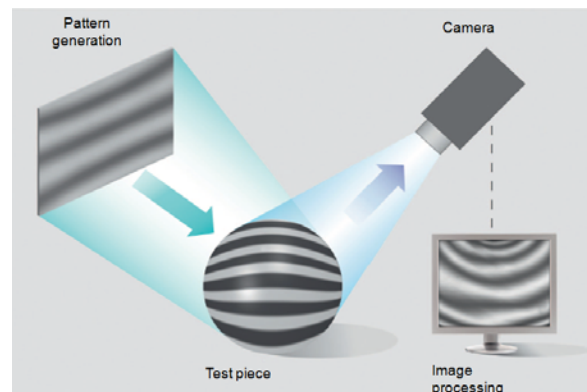


FIGURE 1. Diagram of the phase deflectometry setup.

The resulting images that are acquired are then used to create three critical images for image processing (FIGURE 2)[2]. The first, a 'synthetic grey image', is a very low noise image created by the synthesis of multiple phase shifted images. This image has homogeneous illumination which is widely independent from the shape of the part being tested. This is critical for the evaluation of border areas on curved surfaces (FIGURE 2a).

The second, a 'gloss image', uses the amplitude of the sinusoidal to determine gloss levels which are independent of the shape of the part under test. This image gives the distinction of gloss and brightness which can determine the gloss or matte-like properties of the part under test (FIGURE 2b).

Finally, an 'inclination image' can be obtained from phase information and used to determine the variance of slope on glossy and matte surfaces. This image can be used to visualize indentations or scratches that are not readily visible in the grey image. Additionally, curvature of the object or defects can be derived from this image providing further analysis of the type of defects observed on the surface. These defects include indentations, scratches, pimples, or other negative and positive defects (FIGURE 2c).

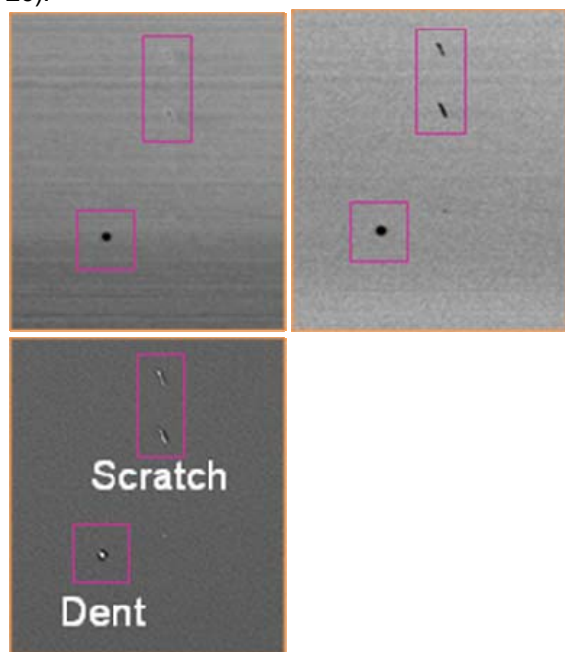


FIGURE 2. a) Synthetic grey image showing a homogenous illumination of the test part. b) Gloss image showing areas which alter the intended gloss level. c) Inclination image showing local defects that have structure in the form of sloped or curved surfaces.

REQUIREMENTS FOR PHASE SHIFTED DEFLECTOMETRY

Phase shifted deflectometry allows for even, non-highly specular surfaces to be realized. This is not the case for typical deflectometry which requires highly specular surfaces (FIGURE 3). Additionally, techniques such as grey scale imaging only work well on matte surfaces.

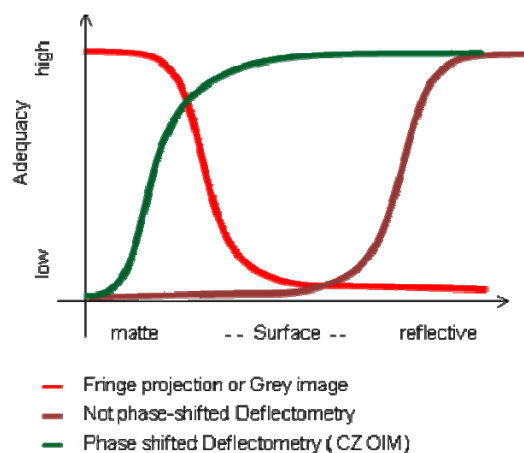


FIGURE 3. Phase shifted deflectometry adequately measures both specular and semi-specular surfaces.

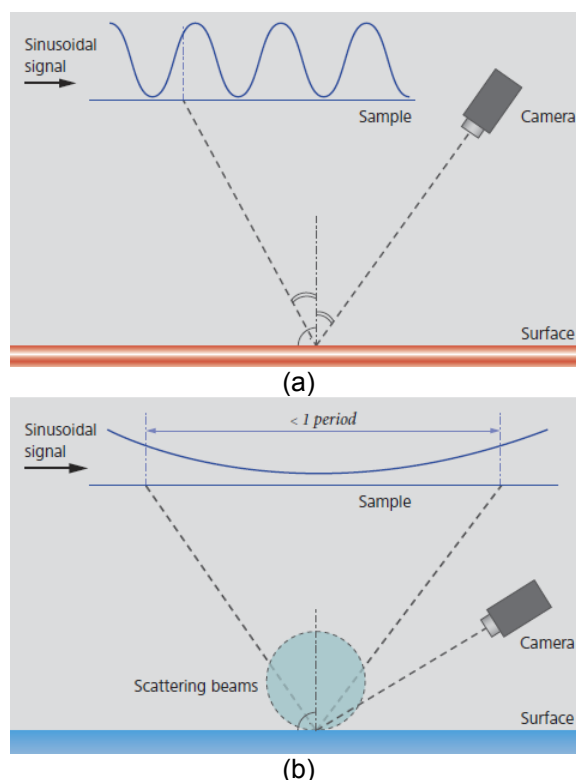


FIGURE 4. a) The use of high sinusoidal fringe patterns is possible on highly specular surfaces. b) As the scattering surface becomes less specular a lower frequency pattern is required to produce less phase uncertainty in the reflection.

As the surface approaches a matte finish and becomes more highly scattering, the phase uncertainty of the reflected pattern increases leading to noisy images. By decreasing the frequency of the projected sinusoidal pattern the uncertainty is removed and these surfaces can

still be imaged for gloss levels and surface defects (FIGURE 4)[3]. This opens up the possibility for a wide variety of surfaces that would not be considered as highly specular to be objectively viewed and evaluated.

RESULTS

Initial imaging results show that this form of a phase measuring deflectometry can objectify many defects that would typically result in non-acceptable parts. Additionally, the observer is able to draw conclusions regarding the severity and nature of the defect allowing for thresholds to be developed and implemented in future test cases, as well as adjust the manufacturing process to correct for unwanted – process related defects.

Minor scratching on a polished free form surface is hardly detectable to the eye (FIGURE 5a) and produces very subtle discrepancies in material height, making it incredibly difficult to sense by touch. With phase shifted deflectometry it is very noticeable using the gloss image that is calculated from the resulting phase images (FIGURE 5b). Additionally, information from the phase channel shows the defect does not create noticeable changes in height and appears to only occur superficially. These results could eventually tell the visual inspector that with some minor spot polishing, this part could be saved from scrapping (FIGURE 5c). This is unlikely to be the case for parts that are more severely scratched.

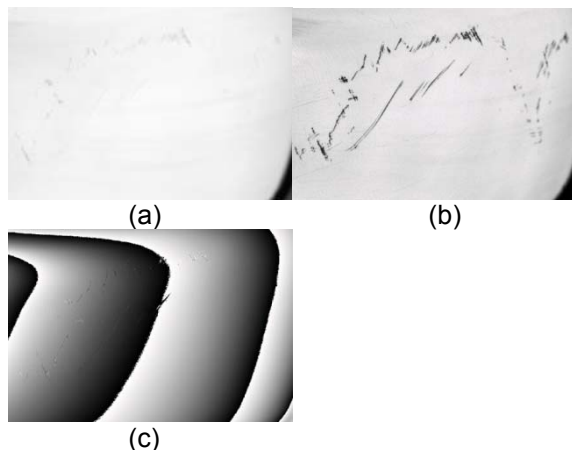


FIGURE 5. a) Grey image, b) Gloss image, and c) Inclination image of subtle surface scratches.

Severe scratches, similar to minor scratches, can be easily missed or misjudged during a visual inspection process (FIGURE 6a). Similar

to minor scratches, these defects are highly visible in the gloss image (FIGURE 6b) but much more serious scratches now show a clear defect to the material in the form of a change in slope and curvature in the phase image (FIGURE 6c). This clarity between severity of the defect can easily help the inspector determine whether a part can be fixed or should be discarded based on the difficulty to fix the defect, saving valuable time and resources during manufacturing.

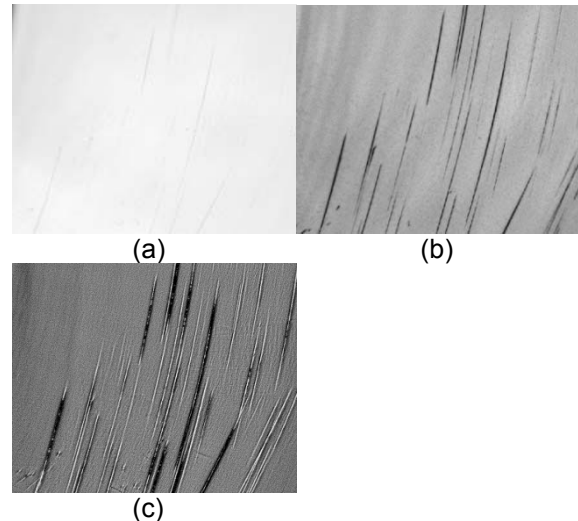


FIGURE 6. a) Grey image, b) Gloss image, and c) Inclination image of severe surface scratches.

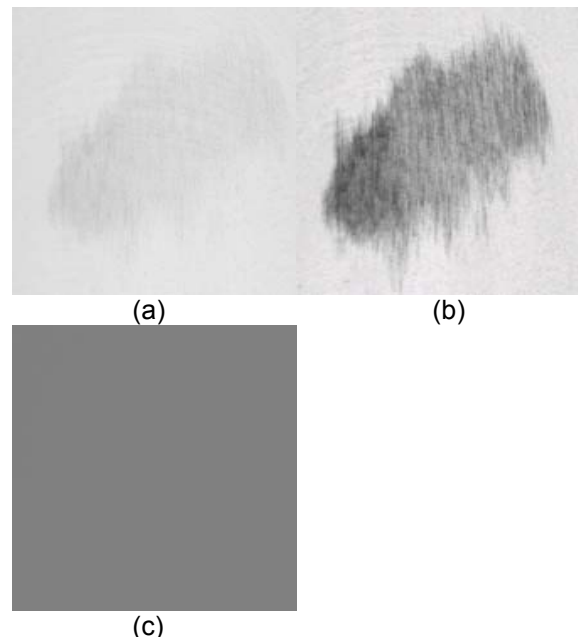


FIGURE 7. a) Grey image, b) Gloss image, and c) Inclination image of surface dull spots.

Perhaps one of the clearest examples of where automated phase shifted deflectometry is superior is in the evaluation of surfaces that have no deviation in surface structure such as a blemish or a dull spot on a finished surface. This very difficult to see defect is barely observable in the gray scale image (FIGURE 7a) and because the defect does not provide any detectable deviation in height from the ideal surface, it goes un-noticed in the phase image (FIGURE 7c). However, with the ability to detect brightness reflections in the gloss image, a unique characteristic of phase shifted deflectometry, the blemish is clearly identifiable and has virtually no chance to escape the inspection process (FIGURE 7b).

Finally, phase shifted deflectometry is able to combine both the gloss and inclination image so that information can be interpolated regarding the manufacturing process. Many times the simple diagnosis that a defect is present does not help solve the problem of why the defect is present.

Simple differentiation of the defect type, identified as a 'nick' or a small removal of material (FIGURE 8a), as opposed to an indentation, can help a manufacturing team focus on upstream processes that are resulting in the defect. Much like the scratch, the 'nick' shows the observer that a defect occurred which has a deviation in local slope and curvature; noticed in the inclination image and similar to what would be expected with an indentation (FIGURE 8c). However, the brightness image is also showing that the surface of the 'nick' is much rougher than the surrounding areas (FIGURE 8b), whereas an indentation would be expected to retain much of the original surface characteristics. Simple separation of these two defect types now enables an observer to note that a manufacturing process such as the polishing of inclusions or dirty polishing elements could be responsible for the defect instead of other common flaws that would normally result in indentations.

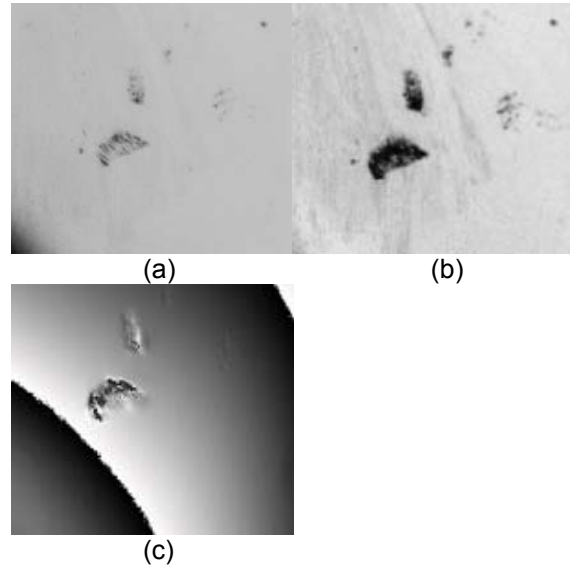


FIGURE 8. a) Grey image, b) Gloss image, and c) Inclination image of surface 'nicks'.

CONCLUSION.

This form of phase shifted deflectometry provides a unique solution to enhancing the capabilities of the visual inspection process. Being able to set objective threshold limits to not only track changes in surface structures but also in gloss levels on free form specular surfaces can greatly increase the consistency and quality that is expected from the visual inspection process. Surfaces that require detailed inspection and high quality control from cosmetic to functional can greatly benefit from this technique. Additionally, manufacturing processes could be optimized based on consistent and reliable feedback of an automated system.

ACKNOWLEDGMENTS

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RECONFIGURATION METHOD FOR OPTIMIZATION OF VOLUMETRIC POSITIONING ERROR OF 3D WORKSPACE

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INTRODUCTION

To ensure the overall motion accuracy of a three-dimensional (3D) workspace, such as a machine tool or coordinate-measuring machine, the volumetric positioning error is used for acceptance and verification of machine performance. As Okafor [1] and Slocum [2] showed, the volumetric positioning error is the different between the actual position of the cutting tool and the workpiece, and their actual positions can be estimated via the Homogenous Transformation Matrix (HTM) using the motion errors of three separate axes including linear, straightness, and angular errors. Many compensation methods have been developed to improve the volumetric positioning error using initial motion errors of each axis and compensated option of controller for designed configuration of machine such as Okafor [1]. However, the volumetric error caused by the motion error of each linear guideway can compensated each other, the optimized configuration can be designed to minimize the volumetric error. Therefore, in this paper, a new method namely Reconfiguration Method is introduced. It is demonstrated that volumetric positioning errors can be improved by using the initial linear guideways with the optimized machine configuration.

RECONFIGURATION METHOD FOR OPTIMIZATION OF VOLUMETRIC ERROR

Volumetric errors are combinations of joint kinematic and link geometric errors of the machine axes. Hence, the volumetric error can be changed by changing the machine configuration or assembly direction of axes (x, y and z-axis). To figure out how these error components effect to volumetric errors, the error maps for the volumetric error are simulated via

Homogenous Transformation Matrix (HTM) with combinations of the kinematic errors and link geometric errors. For the 3-axis machine, there are total 21 geometric errors including 18 errors of 3-axis and 3 squareness errors [1], [2]. All the motion errors of each linear axis and relative parameters between two axes are illustrated in Fig.1 and Fig.2.

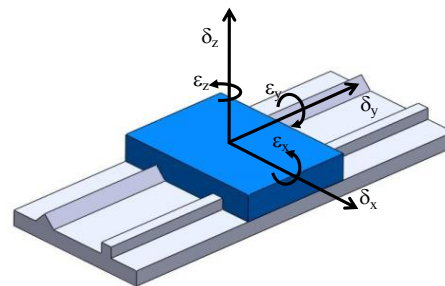


FIGURE 1. Motion errors of linear axis

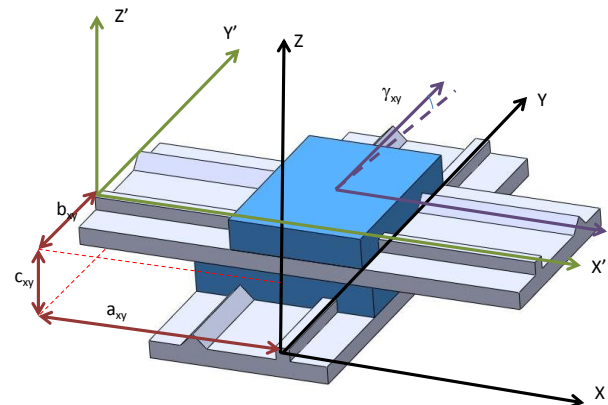


FIGURE 2. Relative parameters

To constitute the volumetric error maps, the HTM matrices must be multiplied to obtain the series of HTMs for tool and workpiece group. To model errors as the machine moves along a specific path, the elements of the HTMs can be numerically assigned values at each point along

the machine's path. The same can generate these positions of the machine's axes. It means that the machine's controller will use to determine where the axes should be relative to each other.

Therefore, the volumetric errors, which represent the translation of toolpoint's coordinate with ideal cutting paths, can be calculated by using below equation.

$$E_{VE} = {}^R T_{tool-real}^{-1} T_{work-real} - {}^R T_{tool-ideal}^{-1} T_{work-ideal} \quad (1)$$

Where: ${}^R T_{tool}$, ${}^R T_{work}$ are series HTMs of cutting tool and workpiece group, respectively.

In case of changing the machine configuration, it can be chosen the optimized configuration depend on minimum deviation of volumetric errors from some possible configurations in the Fig.3. In the Fig.3, the blue and red lines are cutting path of machine with 21 errors and without error, respectively. The difference between the red one and the blue on is the volumetric errors. In the four examples, the volumetric in Fig.3(c) is smallest. Therefore, the configuration XYFZ in Fig.3(c) can be a candidate for optimization of volumetric errors.

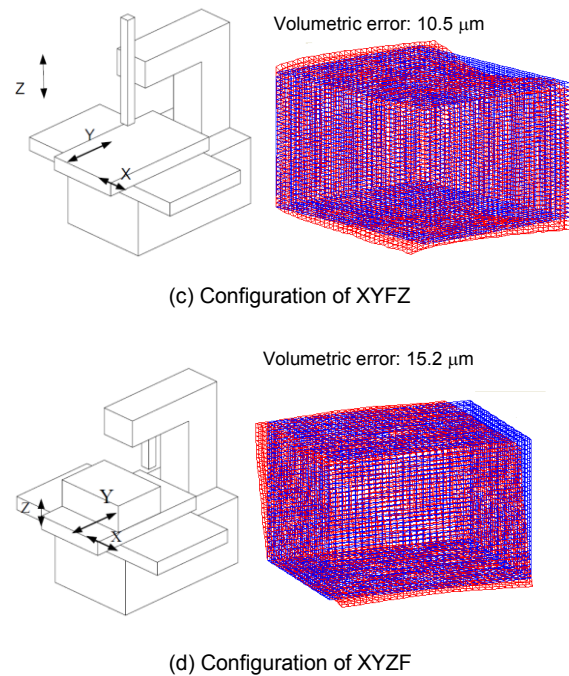
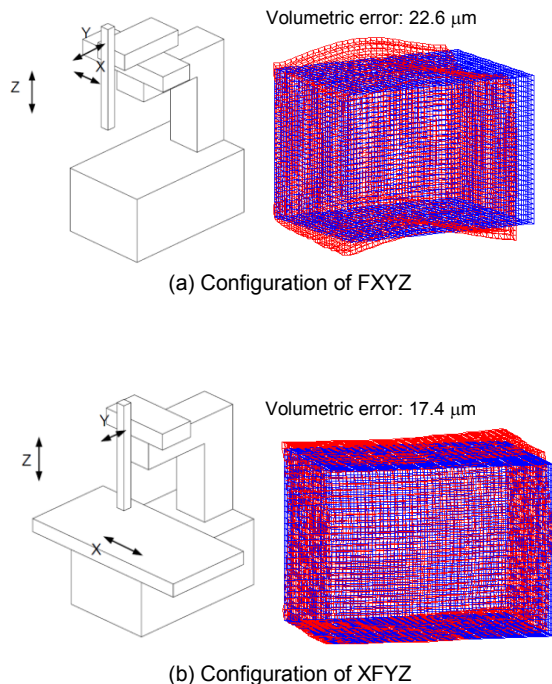


FIGURE 3. Configurations and Error map

In case of changing the assembly direction of axes, the six errors of each axis will not change but the effected direction to the volumetric errors of some errors will change in the machine coordination. Hence, the component errors can increase or decrease volumetric errors by compensation each other in the working volume.

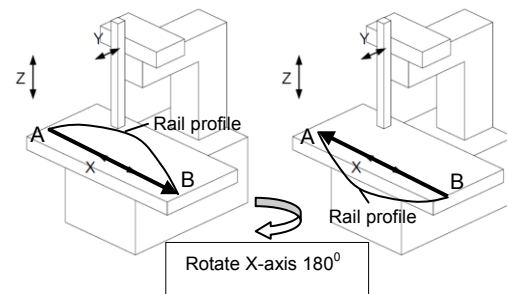


FIGURE.4. Changing the assembly direction

As Fig.4, when the x-axis is rotated 180° around z-axis, the effected direction of six motion errors (linear, vertical and horizontal straightness, pitch, yaw and roll) will change as follow. The errors which effect to x and y direction of 3D working volume (linear, pitch, roll, yaw and horizontal straightness error) will change the effected direction. Otherwise, the error, which effects to z direction (vertical straightness error), will not

change the effected direction. The Fig.5 illustrates the effected directions of the six errors when the x-axis is rotated 180° around z-axis.

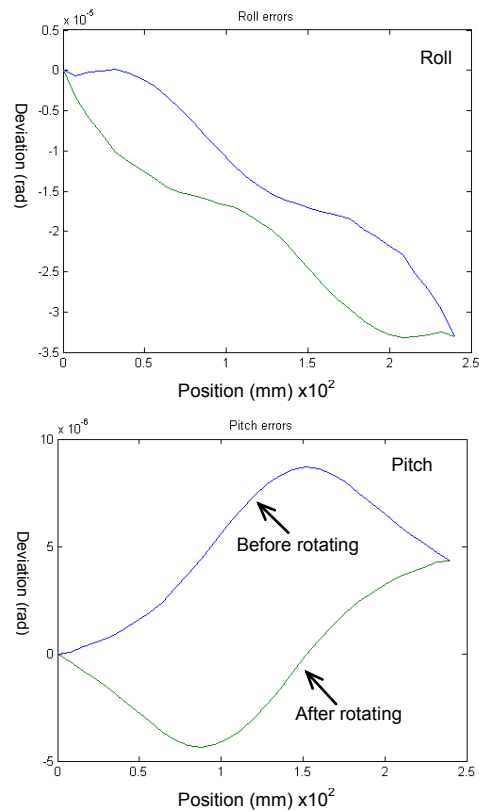
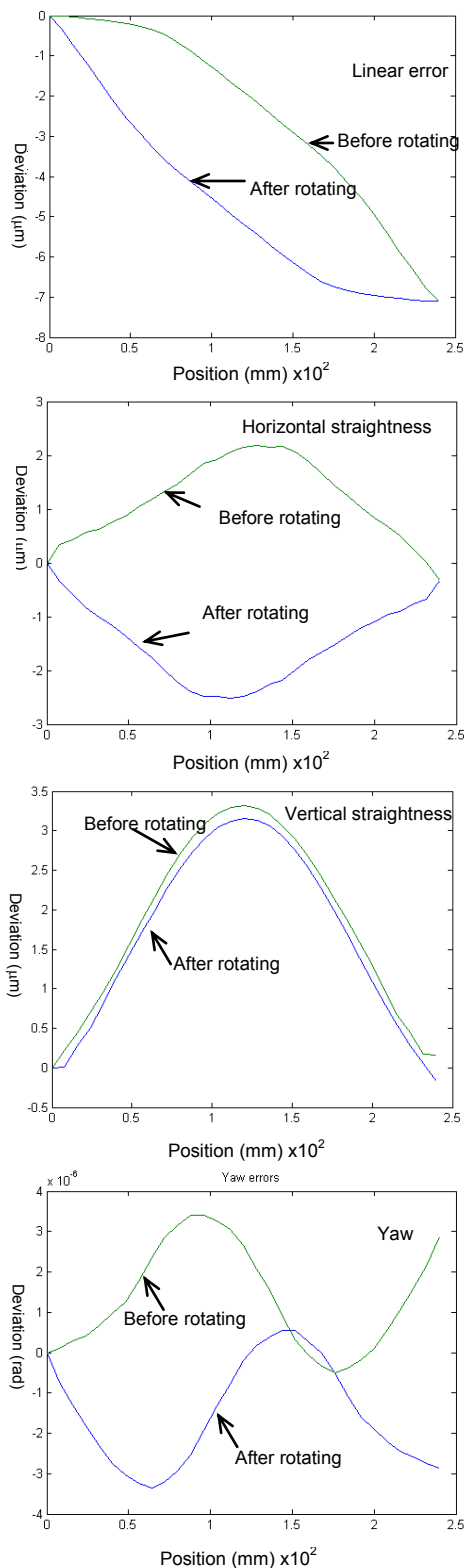


FIGURE 5. Effected directions of six errors

To figure out the effect of changing assembly direction, the machine configuration XYFZ in Fig.3 is used as simulation modeling. From Fig.6 (a), we assume that the volumetric errors can be improved by changing the effected direction following z direction of working volume. Hence, the simulated volumetric errors in machine configuration XYF-Z from Fig.6 (b) (when z was rotated 180°) is improved from 10.5 μm to 5.6 μm .

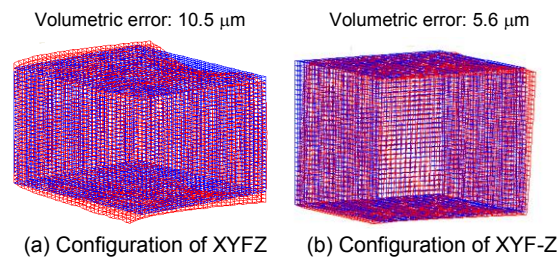


FIGURE 6. Effect of changing assembly directions

CONCLUSION

The results from simulation demonstrate that we can choose the optimized volumetric error for machine configuration by using both changing the machine configuration and changing the effected direction of motion error. By using the changing machine configuration and changing effected direction of the same x, y and z-axis, the volumetric error can be improved from maximum case 22.6 μm to minimum case 5.6 μm .

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INSERTION OF THE HOHLRAUM INTO THE TMP-SHELL: INCREASING DETERMINISM, YIELD, AND THROUGHPUT*

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ABSTRACT

With the NIF ramping up its shot schedule, target assembly is growing away from its early days of one-off prototype targets into a streamlined process that can efficiently produce enough high quality targets. Insertion of the hohlraum into the thermo-mechanical package (TMP) shell, is an example of a critical step in the assembly process that was time intensive and yielded highly variable results. The delicate aluminum TMP shell and gold hohlraum have a very close radial fit of 3.5 μ m, requiring accurate and repeatable alignment for successful insertion.

In order to increase the determinism, yield, and throughput of the hohlraum insertion process, several improvements were implemented. Building on the Hohlraum Insertion Station (HIS) design, a new machine, HIS 2, was designed so that the alignment, insertion, and gluing could be performed on an optical coordinate measuring machine (OCMM), increasing the throughput and determinism of the insertion process. The HIS 2 incorporated many other features to improve the process such as locating adjustment stages to minimize parasitic motions,

increased accessibility to gluing locations, improved kinematic coupling, and redesigning the TMP mounting chuck larger for ease of handling.

This poster presents the HIS-2 design and features for improved hohlraum insertion. It also presents the results of the machine's implementation and its impact on the target assembly process. The poster will also touch on future goals of a 3rd Generation HIS.

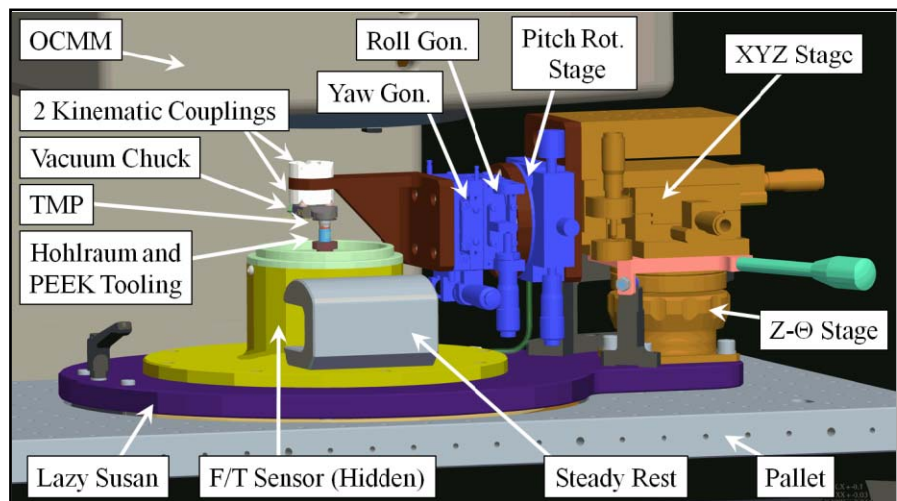


FIGURE 1. ISOMETRIC VIEW
OF HIS CAD MODEL.

*This work performed under the auspices of the U.S. Department of Energy by Lawrence Livermore National Laboratory under Contract DE-AC52-07NA27344.

DEVELOPMENT RESULT OF DOUBLE ARCH-TYPE 5-AXIS MACHINE TOOL

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INSTRUCTIONS

Micromachining tools need to cut harder materials and needs high accuracy. At this trend, machining tools need more powerful motors and accurate control methods. So, the exact estimation of the motion of machine tool is needed.

Also the needs of miniature component markets are increased according to the growth of IT and BT industry. While the components are various, the number is small at this trend. To react quickly to rapidly changing situations, reconfigurable multi-axis machine systems are needed to machine these parts[1-7].

We already developed 1st version of 3-DOF(Degree Of Freedom) double arch-type desk-top machine tool, but we need to enhance the motor capacities and the accuracy. So, we will develop dynamic modeling more simply and more accurate for 5-DOF double arch-type desk-top machine tool as an advanced version. So we did the manipulability analysis of the new machine and dynamic analysis.

The result of simulation and the development result of machine will be present. In next section, the result of the forward kinematics is described. Through the kinematics, the jacobian calculation for manipulability analysis and workspace analysis can be calculated. The development result of machine is presented at the final section.

STRUCTURAL CONFIGURAITON AND KINEMATIC ANALYSIS

The configuration of a double arch-type desk-top machine tool is as follow. Upper part consists of two rotating vertical circular columns, main spindle which moves linearly, and main platform to connect these components. Lower part consists of two planar stage and these make the machine 5 DOF.

In this chapter, the analysis of the machine is focused and the coordinate is defined firstly. The coordinates of this double arch-type machine tool are as follow. The base frame is {B}, the coordinate of the station to put the component on {S}, the position and rotation of tool is {T}, and the goal coordinate to machine component is {G}.

The rotation matrix is as follow.

$${}^B_T = {}^B_S {}^S_T = {}^S_T^{-1} {}^S_T \quad (1)$$

The inverse kinematics is calculated by this relationship between the position and rotation of the main spindle and the rotation of the motors. The position and rotation of the main spindle are calculated firstly to solve this problem.

WORKSPACE ANALYSIS

The workspace of a double arch-type desk-top machine tool is mainly defined by the available rotational angles. The rotational angles are restricted by the collision between the vertical column and main spindle housing. So we made

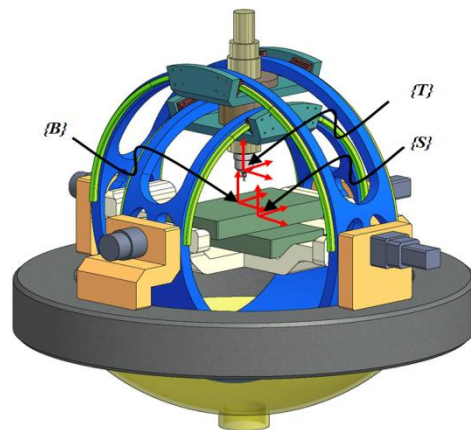


FIGURE 1. The configuration of a double arch-type machine tool.

the groove at the guides to secure an enough workspace. The workspace is $-60^\circ \sim +60^\circ$.

JACOBIAN ANALYSIS

Jacobian is defined as the linear relationship between the time derivatives of related variables. The forward Jacobian can be solved to define the relationship between the velocity of the tool platform and the time derivative of the actuating joints. Manipulability analysis and singularity analysis can be achieved by using the forward jacobian.

First, define the platform linear velocity, v can be calculated by differentiating the position vector, P_t of platform by time as follows.

$$v = \frac{dP_t}{dt} = \frac{dP_t}{dq} \frac{dq}{dt} = J_v \dot{q} \quad (2)$$

When the rotation matrix R is given, the angular velocity ω is can be calculated as bellows.

$$\hat{\omega} = R^T \dot{R} \quad (3)$$

where,

$$\omega = \begin{bmatrix} \omega_1 \\ \omega_2 \\ \omega_3 \end{bmatrix}, \quad \hat{\omega} = \begin{bmatrix} 0 & -\omega_3 & \omega_2 \\ \omega_3 & 0 & \omega_1 \\ -\omega_2 & -\omega_1 & 0 \end{bmatrix}.$$

So, the rotation matrix R is defined as above, $\omega_1, \omega_2, \omega_3$ is as follows.

$$\omega_1 = R_z^T \dot{R}_y, \quad \omega_2 = -R_z^T \dot{R}_x, \quad \omega_3 = -R_x^T \dot{R}_y. \quad (4)$$

So, the rotation matrix R is defined as above, $\omega_1, \omega_2, \omega_3$ is as follows.

$$\omega = \begin{bmatrix} \omega_1 \\ \omega_2 \\ \omega_3 \end{bmatrix} = \begin{bmatrix} c\beta c\alpha \dot{\gamma} + s\alpha \dot{\beta} \\ -c\beta s\alpha \dot{\gamma} + c\alpha \dot{\beta} \\ s\beta \dot{\gamma} + \dot{\alpha} \end{bmatrix} \quad (5)$$

By using this relationship, the angular velocity of the platform can be calculated by the relationship of jacobian J_ω and joint angular velocity $\dot{q}$.

$$\omega = J_\omega \dot{q} \quad (6)$$

The forward jacobian is defined include both the platform velocity and the angular velocity as below.

$$\dot{P}_t = \begin{bmatrix} v \\ \omega \end{bmatrix} = \begin{bmatrix} J_v \\ J_\omega \end{bmatrix} \dot{q} = J \dot{q}, \quad J \in \mathbb{R}^{6 \times 3}. \quad (7)$$

The linear acceleration and the angular acceleration can be calculated as below.

$$\dot{v} = \dot{J}_v \dot{q} + J_v \ddot{q}, \quad (8)$$

$$\dot{\omega} = \dot{J}_\omega \dot{q} + J_\omega \ddot{q}. \quad (9)$$

MANIPULABILITY ANALYSIS

Manipulability is the transfer characteristic between input and output of the velocity and the torque. That is, the ability to move to a specific direction, or transferring ability of the forces. It is important to design manipulators with enhancing manipulability.

The velocity manipulability can be calculated by using the forward jacobian from equation, $\dot{P}_t = J \dot{q}$. If the angular velocity is given as unit circle ($\|\dot{q}\|^2 = 1$), the velocity of tool platform can be calculated and the shape is ellipsoid. This called manipulability ellipsoid, and the major axis and minor axis can get from the maximum and minimum eigenvalue and eigenvector of JJ^T . When maximum and minimum eigenvalue of

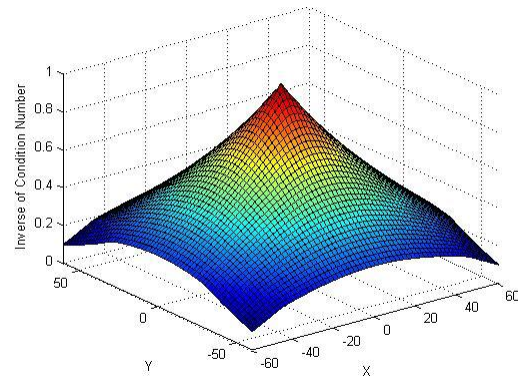


FIGURE 2. The manipulability graph inside the workspace

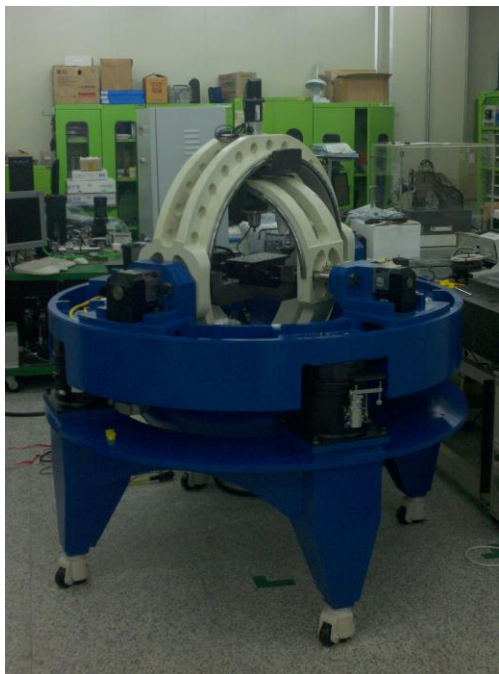


FIGURE 3. Whole system of multi-axis reconfigurable system

JJ^T is λ_1, λ_2 ($\lambda_1 > \lambda_2$), and the eigenvectors are v_1 and v_2 , the major axis is v_1 whose size is $\sqrt{\lambda_1}$ and minor axis is v_2 whose size is $\sqrt{\lambda_2}$. To make the characteristic of input and output uniform, the manipulability ellipsoid should become circle shape. The distortion of ellipsoid is smaller, the manipulability is better.

This manipulability can be show as the condition number of forward jacobian quantitatively.

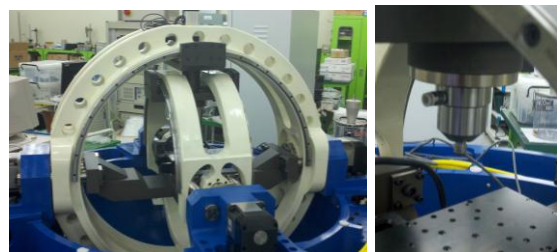
$$\text{Manipulability measure } c(f) = \lambda_{\max} / \lambda_{\min}. \quad (10)$$

Where, the number is smaller, the ellipsoid is more similar to circle and manipulability is better.

Then we calculate the manipulability inside the workspace and the result is shown at figure 2. Manipulability of this manipulator is good near the home position, $(0^\circ, 0^\circ)$. Then the larger the angle is, the worse manipulability is. However, there is not singularity inside the workspace.

DEVELOPMENT OF THE MACHINE TOOL

Manufactured multi-axis machining system is organized as follows. Two rotating vertical column rotates by 2 motors in the condition of synchronization control. The spindle motor can



(a) Vertical circular columns (b) XY stage
FIGURE 4. Whole system of multi-axis reconfigurable system

be moved up or down to the spindle axis for the linear motion unit and 2-axis stage is composed of a planar one.

Based on the above configuration of the machine was manufactured. The machine is divided largely into the machine body and controls. Mechanism stage is divided as the vertical circular column, spindle part, and a planar stage. The shape of each part is as shown at Figure 3 and Figure 4.

CONCLUSIONS

A double arch-type of 5-axis machining tool has been developed. This machine has an advantage of spherical coordinates motion by intersecting two vertical columns at one point.

As a future work, measuring the accuracy and maximum speed of system to verify the performance of system will be present.

ACKNOWLEDGEMENT

This work was financially supported by the grant from the Industrial Source Technology Development Program of the Ministry of Knowledge Economy(MKE) of South Korea.

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PERFORMANCE ANALYSIS OF ACTIVE AEROSTATIC THRUST BEARINGS WITH DYNAMIC ERROR BUDGETING

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INTRODUCTION

Mechatronics has allowed for extended functionalities and improved performances in the field of precision engineering, but on the expense of higher design complexity and more disturbance sources. Machine design needs to deal with including sensors and actuators, finding their optimal type and location, defining the control logic and the electronics on which to implement it, etc. These elements contribute to the performance of precision machines, but also add error sources or limitations that need to be considered, such as the noise level of the sensors which limits the measurement resolution. Mechatronics has also allowed for faster machines, increasing the relevance of considering and optimizing the dynamic characteristics of the machines.

Optimal design of precision mechatronic devices requires the use of tools adapted from precision and control engineering communities. Precision engineering tools, such as error budgeting, are prepared for considering the effect of multiple disturbances. Control engineering tools, e.g. based on Bode diagrams, can deal with feedback control systems and dynamic performance of the system. Dynamic Error Budgeting (DEB) can meet these requirements, combining and extending classical error budgeting and control design and analysis based on Bode diagrams.

Active aerostatic thrust bearings are a good example of the new opportunities and challenges brought by mechatronics into precision engineering, adding active position control capabilities to air bearings.

The performance of active air bearings has been analyzed experimentally and compared with predictions based on DEB. The results presented here demonstrate the potential of dynamic error budgeting and active aerostatic bearings for precision mechatronics.

ACTIVE AEROSTATIC THRUST BEARINGS

Aerostatic bearings are commonly used in precision engineering machines, such as coordinate measuring machines or wafer steppers. They enable highest precision and resolution motion, but at the expense of relatively lower stiffness and damping compared to contact or hydrostatic bearings [1].

Active control comes as a solution to overcome these intrinsic limitations and extend their functionality by adding short stroke positioning capability. A number of alternatives have been proposed in the literature, from feed pressure control [2] to the use of piezoelectric actuators embedded in the bearing, either around the restrictor [3] or on the periphery of the bearing surface [2].

The last approach is followed here, since it is considered to offer the best performance in terms of resolution and control bandwidth. Three piezoelectric actuators are embedded in the bearing on its periphery, acting on a flexible bearing pad clamped on a central column [4] (see FIGURE 1). The elongation of the actuators produces a change in the shape of the bearing surface, and thus a change in the pressure distribution and in the bearing force.

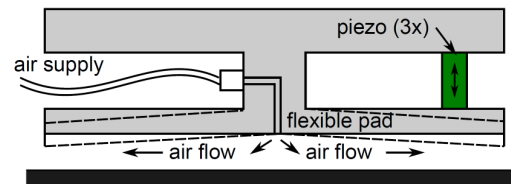


FIGURE 1. Sketch and working principle of an active aerostatic bearing

The performance of the active air bearings (stiffness, positioning resolution, bandwidth) is affected by a number of factors of different nature, e.g. the design of the active air bearing

itself, broadband disturbances such as the electronic noise in the sensors and actuators, digital data processing, oscillations in feeding pressure, vibrations coming from the ground, etc. The achievable resolution is in the nanometer range, and any of these disturbances can be the factor limiting the performance. The design and performance of such systems can be thus analyzed by DEB.

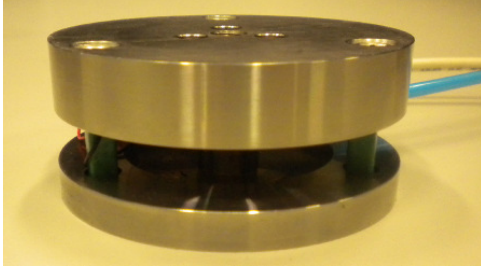


FIGURE 2. Active aerostatic bearing prototype

DYNAMIC ERROR BUDGETING

General Theory

The ever-increasing precision requirements in positioning and guiding systems have demanded the development of design tools for predicting and analyzing their performance. Error budgeting is a classical approach in precision engineering for allocating the allowed error within all the components of the system. Dynamic Error Budgeting [5] (DEB) has been proposed for precision mechatronic systems, where the influence of the frequency-dependent disturbances acting on a closed loop system needs to be analyzed too. DEB allows the designer to assess the influence of each of the components of the active system and improve the performance over cost ratio by reducing development times. It can also be used on an existing system in order to find the limiting factor where improvement efforts should be focused.

In DEB, the disturbances can be modeled as stationary stochastic signals, and their propagation through the control loop is analyzed in a systematic way by using their Power Spectral Density (PSD), and the system plants are represented by their Frequency Response Function (FRF). The analysis is thus based on the propagation of uncorrelated stochastic signals through transfer functions.

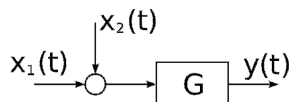


FIGURE 3. Propagation of x_1 and x_2 through G

As an example, in FIGURE 3, the addition of two disturbances is easily represented by their PSD:

$$S_x = S_{x_1} + S_{x_2} \quad (1)$$

The effect of a transfer function on the signal is calculated as follows:

$$S_y(j\omega) = |G(j\omega)|^2 S_x(j\omega) \quad (2)$$

The validity of the method relies on how accurately these PSD and FRF fit the reality. Modeling and identification are discussed in detail in [5].

The outcome of the DEB is the Cumulative Amplitude Spectrum (CAS) of the total error of the system, and the contribution of each of the disturbances (see FIGURE 9). The end value of the CAS is the total error, and this value can be used to assess and optimize the performance of the active system (see FIGURE 7).

DEB for Active Aerostatic Bearings

The closed loop system considered for the control of active air bearings is represented in FIGURE 4. K is the controller, FF is the feedforward plant, P is the active bearing (displacement over voltage) and P_d is the disturbance plant, r is the tracking reference, sn is the sensor noise, f the external noise, fl is the floor vibration and ac is the actuator noise. The real error e and the measured error e_m are calculated as follows:

$$\begin{aligned} e &= r - y = S((I - FF)r - Pac - fl - P_d f) + (I + S)sn \\ e_m &= e - sn = S((I - FF)r - Pac - fl - P_d f - sn) \end{aligned} \quad (3)$$

where $S = (I - PK)^{-1}$ is the control sensitivity function.

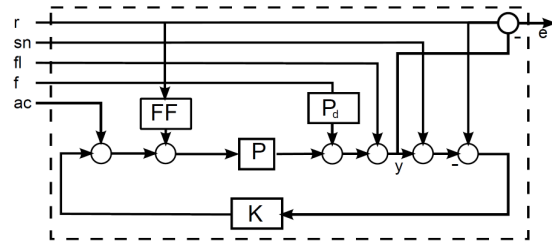


FIGURE 4. Closed loop control system with main disturbances

All the elements of the system need to be modeled based on experiments, catalogue data

or theoretical models. The main disturbances considered here are discussed next.

Disturbance and plant modeling

- **Transducer noise**

Capacitive sensors and piezoelectric actuators operate based on an electric signal that is treated by electronic equipment for amplification, linearization, etc., and there is thus a direct relation between the electric noise generated by this equipment and the measurement or displacement that are obtained.

This disturbance is usually expressed in terms of signal-to-noise ratio, and in practical terms it means that the larger the required measurement and actuation range needed, the larger the positioning uncertainty will be.

- **Digital signal processing**

Quantization (digital to analog (DA) and analog to digital (AD)) and discretization of the transducer signals can also affect the positioning resolution and bandwidth of the active system. These effects are typically non-linear, so the proposed models [5] are not so well represented by the frequency domain analysis proposed here. In case DEB predicts that any of these effects is dominant, a time domain simulation would be needed for accurate results.

- **External forces**

The active bearing will be affected by external effects, such as floor vibrations, forces coming from inertial effects or a machining process, and also the oscillations of the feed pressure. Since these effects are very variable from case to case, their analysis is usually based on experimental data.

- **Position reference**

The variation of position reference can be considered as a disturbance, since it introduces an error to the system that needs to be compensated by the control. When the tracking signal is known, it can be modeled in DEB by its PSD.

EXPERIMENTAL VALIDATION

Test Setup

The DEB and active control method proposed here has been tested experimentally on a setup that allows static and dynamic characterization of single active aerostatic thrust bearings (see FIGURE 5). An electrodynamic shaker is used to apply disturbance forces, three Lion Precision

capacitive sensors are used to measure the displacement of the bearing, and a Kistler sensor is used to measure the force. A DSPACE 1104 real time control system is used for data processing.

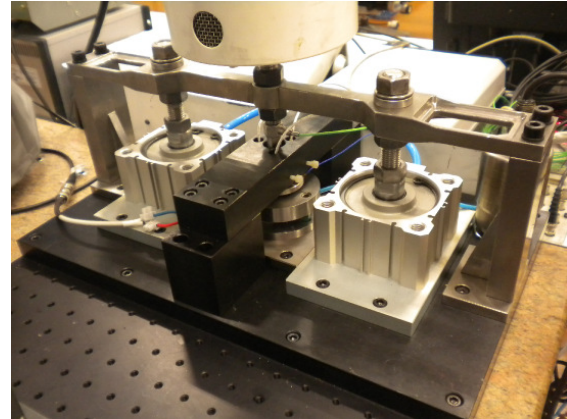


FIGURE 5. Active air bearing test setup

Plant identification

The test setup has been used first for the characterization of the system plants: P , P_d and FF . The control plant P and the disturbance plant P_d are shown next:

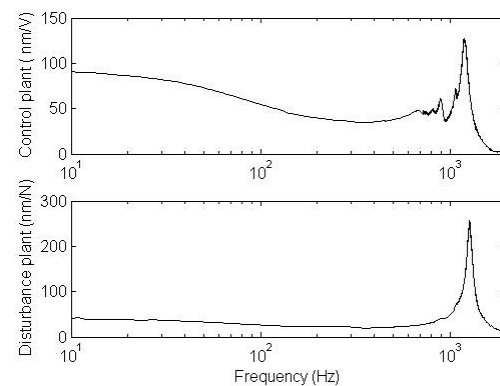


FIGURE 6. Control P and disturbance P_d plants

Controller

The DEB method as proposed here can be used with any controller that can be expressed as a linear transfer function in frequency domain. For simplicity, plain integral control is used here, and the performance of the system for different integral gains will be analyzed in the following experimental results.

Disturbance rejection

The shaker can be used to analyze the effect of external disturbance forces. In this case, a force

of 10 N rms is applied, distributed between 1 and 50 Hz. The total error for different controller gains is shown in FIGURE 6. On the one hand, the feasibility of almost eliminating the vibration produced by external forces with the active control is demonstrated. On the other hand, the DEB analysis is able to predict reasonably well the vibration that is produced in each case.

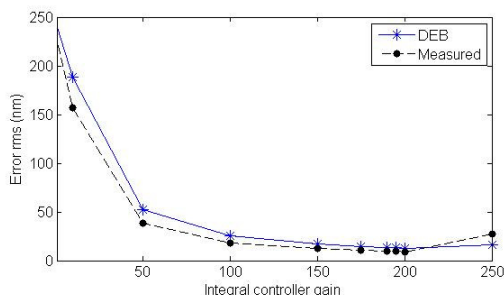


FIGURE 7. *Vibration produced by external forces for different controllers*

Free vibration

When there are no external forces applied to the bearing, there is in principle no clear dominant disturbance as in the previous case. In FIGURE 9 the total measured vibration level is shown for different controllers. The DEB analysis is able to predict with good accuracy the vibration level between the bearing and its counter surface and the influence of the controller.

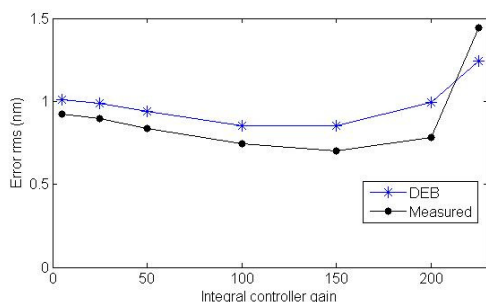


FIGURE 8. *Bearing vibration under no external forces for different controllers.*

In FIGURE 8, the case of the controller gain set to 150 is analyzed in more detail, showing the CAS of the disturbances and the total error. We can observe that the floor vibration is eliminated by the controller, and the sensor noise is the dominant disturbance.

CONCLUSIONS

Dynamic error budgeting has been proposed as a tool for analyzing precision mechatronics systems, and its application to active air bearings has been presented.

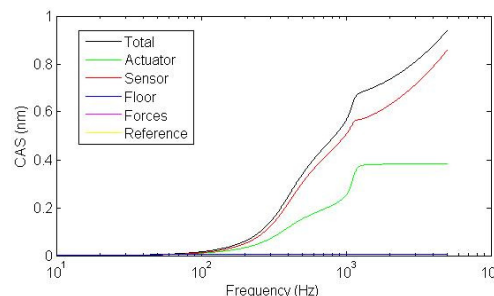


FIGURE 9. *CAS of the vibration of the bearing and the contribution of each disturbance*

The experimental results demonstrate the high performance that can be achieved by active air bearings, in this case keeping the vibration level at nanometer level even in presence of relevant external forces.

The results have been analyzed by DEB method, which has proven itself as a useful tool for assessing the performance of high precision mechatronic systems affected by different disturbance sources.

ACKNOWLEDGEMENT

Some concepts presented here were partially developed during the PhD research work of Gorka Aguirre at the K.U. Leuven (Belgium), under the supervision of Prof. H. Van Brussel and Prof. F. Al-Bender.

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DESIGN OF A DETERMINISTIC FIXTURE FOR HANDLING CFTA'S USED IN INERTIAL CONFINEMENT FUSION TARGETS

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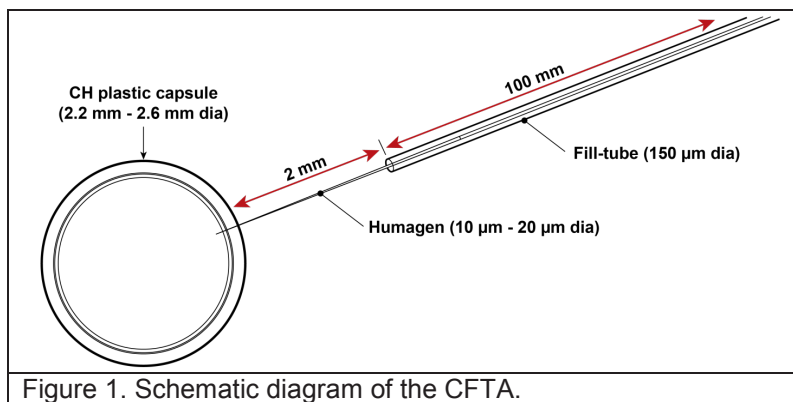
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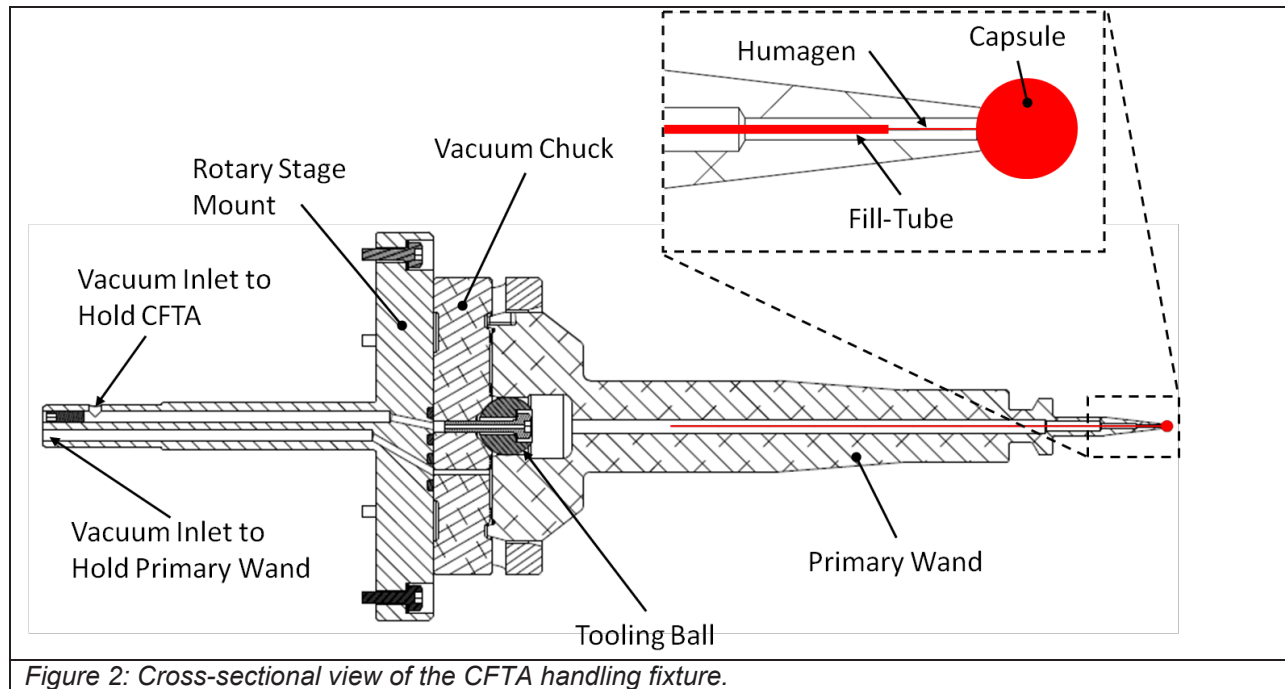
This abstract discusses the design of a deterministic fixture to handle capsule fill-tube assemblies (CFTA) used in Inertial Confinement Fusion (ICF) targets on the National Ignition Facility (NIF)[1]. The CFTA, see Figure 1, consists of a hollow Ge-doped plastic sphere, called a capsule, ranging in outer diameter between 2.2 mm and 2.6 mm and an attached 150 μm diameter glass-core fill-tube that tapers down to a 10 μm diameter at the capsule.

A fixture, see Figure 2, was designed to manipulate the CFTA such that the entire surface of the capsule can be imaged using the 3D Surface Mapping System (3D SMS) [2], while providing a deterministic mounting better than 1 μm .

A detailed discussion of the design of the vacuum chuck and primary wand will be presented, which will include requirements, system functionality, contact mechanics and transport method.

This work performed under the auspices of the U.S. Department of Energy by Lawrence Livermore National Laboratory under Contract DE-AC52-07NA27344. LLNL-ABS-538171





[1] <https://lasers.llnl.gov/>

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PRECISION MEASUREMENT OF NUT STIFFNESS IN BALLSCREWS

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ABSTRACT

To verify previously developed nut stiffness model of a ballscrew and compensate for tilting of the ballscrew shaft with respect to the nut body, an improved stiffness measurement system has been devised. Double-nut ballscrews are tested under several load and assembly conditions. It is confirmed that experimental results are around 90% of the mathematical nut stiffness.

INTRODUCTION

Ballscrews are common positioning units applied to industrial machinery and precision machines. As the positioning accuracy of a ballscrew driven system depends upon stiffness of the leadscrew system, stiffness estimation of the ballscrew is critical for accurate positioning and design of the precision machinery.

Fig. 1 shows cross sections of a double-nut ballscrew in TC and TT load conditions. Contact state is shown in Fig. 1(b).

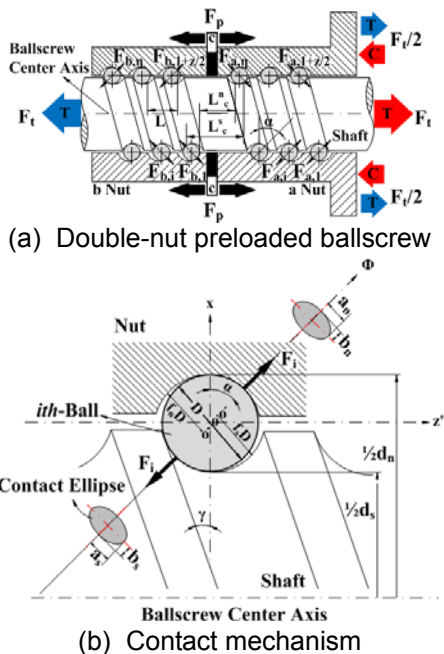


FIGURE 1. Ballscrew and contact mechanism.

According to previous research [1-2], Hertzian deformation constants on the ball and grooves of a ballscrew is given by

$$C_{bs,bn} = \frac{\delta_{bs,bn}^* \sum \rho_{bs,bn}}{2} \cdot \left[\frac{3}{2 \sum \rho_{bs,bn}} \left(\frac{(1-\zeta_b^2)}{E_b} + \frac{(1-\zeta_{n,s}^2)}{E_{n,s}} \right) \right]^{\frac{2}{3}} \quad (1)$$

Hertzian deformation of nut *a* and *b* parts according to preload and thrust force is given by

$$\delta_t^H = \left(\delta_{p+t}^{A,H} + \delta_{p+t}^{B,H} \right) / 2 \quad (2)$$

Body deformation of nut and shaft with *p* balls case is given by

$$\delta_{total}^B = - \sum_{i=2}^p \delta_{i-1,i}^s \pm \sum_{i=2}^p \delta_{i-1,i}^n \quad (3)$$

Then, total deformation due to Hertzian and body deformation is given by

$$\delta_{total} = \delta_t^H + \delta_{total}^B \quad (4)$$

Ballscrew stiffness according to thrust force *F_t* is given by

$$S_t = F_t / \delta_{total} \quad (5)$$

Fig. 2 shows contact load distribution of DFT2505-5 (NSK's double-nut ballscrew) for TT(CC) and TC(CT) load cases when preload is 129Kgf. Using the computed contact load and Eqs. (5), axial deformation and stiffness of DFT2505-5 according to thrust force when preload is 129Kgf are computed as shown in Fig. 3. For TT or CC load case when *F_t* = 200Kgf and 700Kgf, stiffnesses

are computed by 87.98 and 84.24 Kg μ m, respectively.

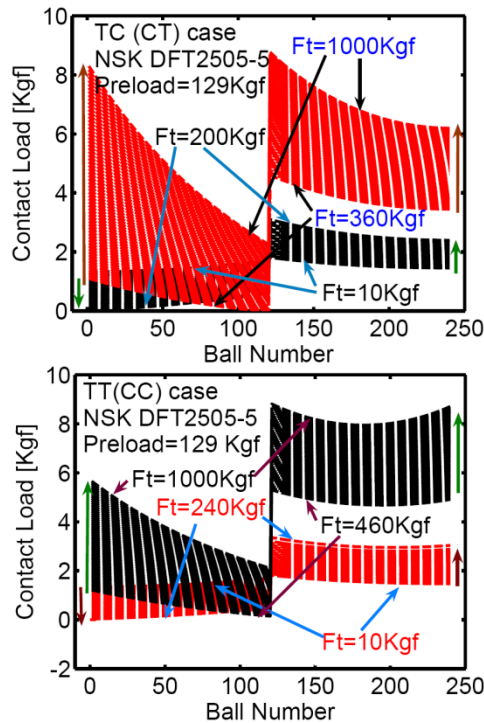


FIGURE 2. Contact Load Distribution of DFT2505-5 with Preload 129Kgf vs. Thrust Force.

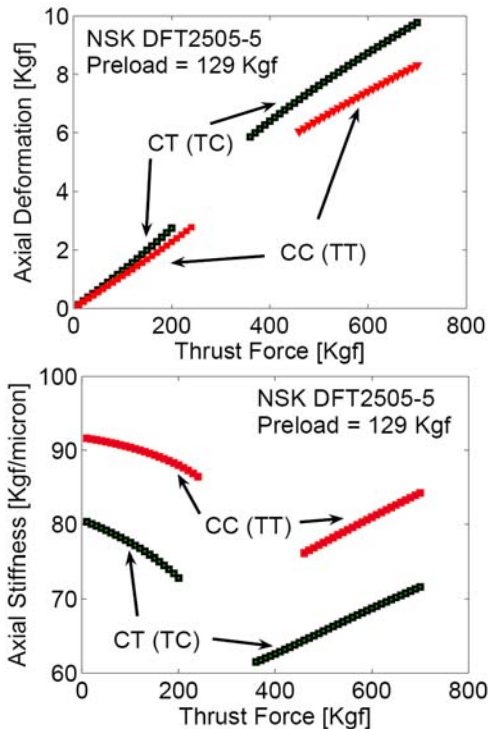


FIGURE 3. Axial Deformation and Stiffness of DFT2505-5 with Preload 129Kgf vs. Thrust Force.

In the previous study [2], experimental setup shown in Fig. 4 was applied for stiffness measurement. Two gap sensors were used for axial deformation measurement. Fig. 5 shows measurement results of stiffness from 2 sensors. During relative deformation measurement, tilting of the ballscrew shaft occurred between the nut and the shaft. Both have different stiffnesses due to the tilting. Uncertainty of the sensors were observed a lot as well. The tilting generates attitude change of the sensor target on the experimental setup. To reduce the measurement uncertainty and tilting during deformation measurement, a compensation method of the shaft tilting and the precision measurement of nut stiffness are required through upgrade of previous measurement system.



FIGURE 4. Previous Experimental Setup.

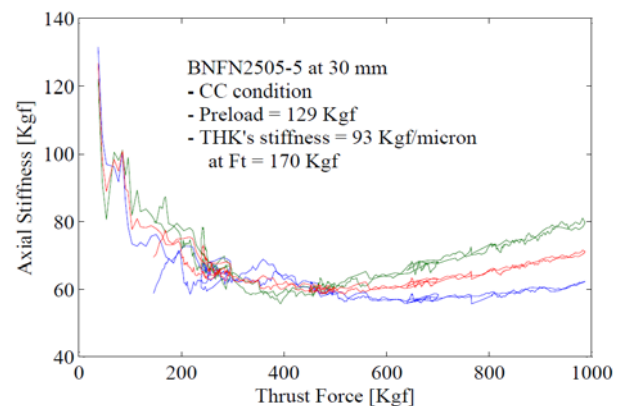


FIGURE 5. Stiffness Measurement in CC Load Condition with 129Kgf Preload.

In this paper, improvement of the experimental setup with 5 sensors to compensate for the tilting and with nano-level accuracy is performed for precision measurement of the ballscrew nut stiffness. Verification of the previously developed mathematical model [1] of the ballscrew nut

stiffness is conducted on the developed stiffness measurement system.

EXPERIMENTS

An improved stiffness measurement system composed of a hydraulic cylinder, 5 LVDTs (resolution 0.01 μ m, TESA GT21HP with TESATRONIC TT80) with tilting compensator, load cell (Bongshin, 2 tons) and data acquisition system (NI, LabView) as shown in Fig. 6 is devised for the stiffness measurement. To compensate for the tilting error, a tilting angle measurement system shown in Fig. 7 is constructed. During loading according to thrust force, the sensor target varies tilting angles θ and ϕ of the ballscrew with respect to x and y axes. These tiltings distort axial relative deformations measured by LVDT A. To compensate for the tilting, tilting angles are identified by trigonometric measurement of B, C, D and E displacements. Then kinematic measurement chain is defined as shown in Fig. 7. After calibration of the tilting error, true axial deformation ΔZ is obtained from Eq. (6).

$${}^{CS1}P_{CS2} = [R_Y] \cdot [R_X] \cdot {}^{CS1}P_{CS2}$$

$$\begin{bmatrix} L+\Delta x \\ 0+\Delta y \\ L_T+\Delta z \\ 1 \end{bmatrix} = \begin{bmatrix} \cos\phi & \sin\phi\sin\theta & \sin\phi\cos\theta & 0 \\ 0 & \cos\theta & -\sin\theta & 0 \\ -\sin\phi & \cos\phi\sin\theta & \cos\phi\cos\theta & 0 \\ 0 & 0 & 0 & 1 \end{bmatrix} \begin{bmatrix} L \\ 0 \\ L_T \\ 1 \end{bmatrix} \quad (6)$$

Corresponding to the double-nut NSK's DFT2505-5 for the simulation, THK's BNFN2505-5 are selected for experimental verification. Fig. 8 (a) shows B,C,D,E LVDT sensor outputs, load cell output, and Z-axis displacement measured by A LVDT. The measurement result confirms tilting during loading. Fig. 8 (b) shows sensor target distortion during loading. After compensating for the measured tilting motion through Eq. (6), axial deformation and stiffness are obtained as shown in Fig. 8 (c) and (d).

During the experiment, preload is fully released at the critical load around 400Kgf thrust force, and the loading continues up to 700Kgf thrust force. Within the critical region, the stiffness decreases according to the load increase. The stiffness increases after passing the critical load. Based on Fig. 3, mathematically obtained axial deformation and stiffness of DFT2505-5 with 129Kgf preload are 2.27 μ m and 87.98Kgf/ μ m at 200Kgf thrust force. Measured deformation and stiffness are 2.35 μ m and 85Kgf/ μ m at 200Kgf thrust force, respectively. Analytical stiffness at 700Kgf is

84.24Kgf/ μ m. Measured stiffness at 700 Kgf is 77Kgf/ μ m. It confirms that experimental stiffness is around 96.73% and 91.4% of the analytical value at 200Kgf and 700Kgf thrust force, respectively. In addition, THK's recommended stiffness value of BNFN 2505-5 at 170Kgf thrust force is 93Kgf/ μ m. It is obtained as 87Kgf/ μ m (93.5% similarity) in the experiment.

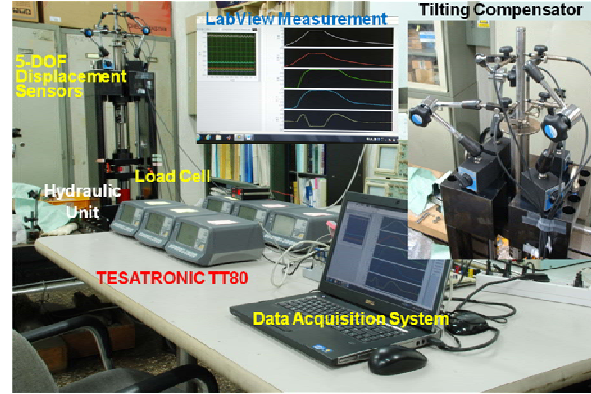


FIGURE 6. Improved Experimental Setup.

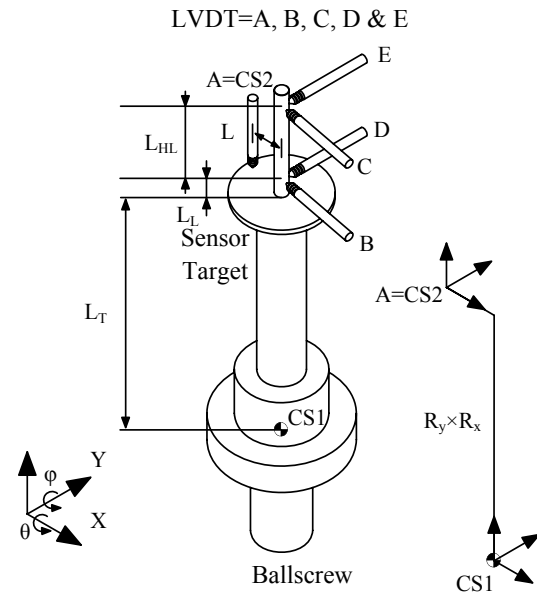
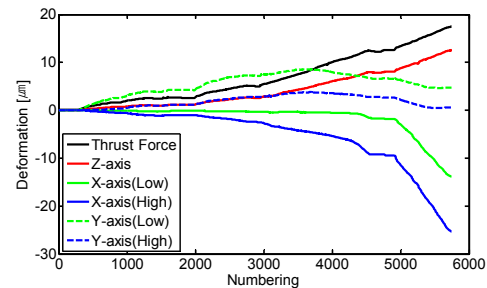
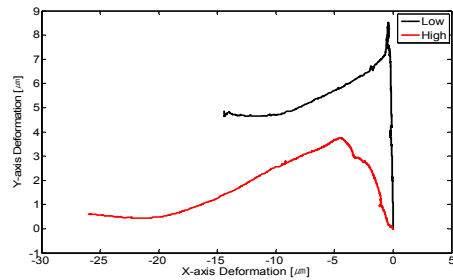


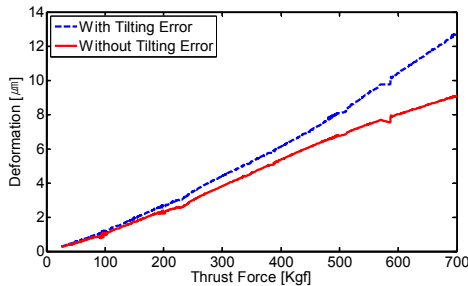
FIGURE 7. Tilting Compensator and Coordinates.



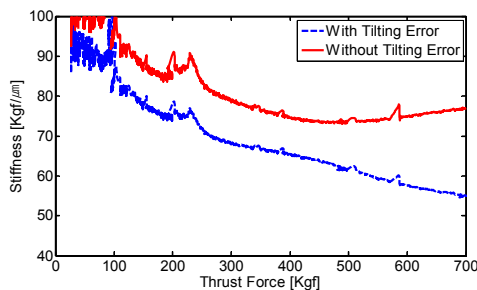
(a) 5 LVDTs and Load Cell Measurement



(b) Tilting Motion in XY-plane



(c) Compensated Axial Deformation



(d) Compensated Axial Stiffness

FIGURE 8. Improved Stiffness Measurement in CC Load Condition with 129Kgf Preload.

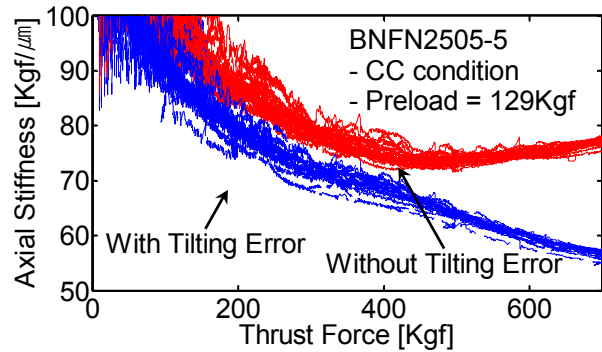
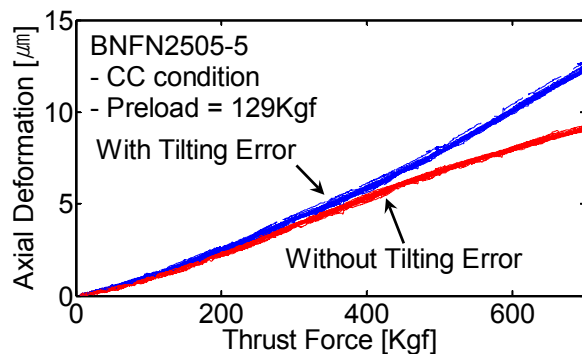


FIGURE 9. Experimental Axial Deformations and Stiffnesses Before and After Compensation.

Fig. 9 shows 30 experimental results of BNFN2505-5 for CC load and 129 Kgf preload cases. Mean deformations at 200 Kgf and 700 Kgf thrust force are 2.24 μ m and 9.30 μ m, respectively. Mean stiffnesses are 89.3Kgf/ μ m (101.5% of the analytical value) and 75.3Kgf/ μ m (89.38% of the analytical value) at 200 Kgf and 700 Kgf thrust force, respectively.

CONCLUSIONS

Experimental verification of the nut stiffness model of ballscrews is performed through improved stiffness measurement system. Calibration of the tilting motion of the ballscrew shaft in the nut during loading process is conducted by using the devised tilting compensator. It is confirmed that theoretical stiffness model has 90% similarity with the experimental results.

ACKNOWLEDGEMENT

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MECHANICAL VACUUM FEEDTHROUGH USING GAS LUBRICATED GUIDING ELEMENTS

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ABSTRACT

Aerostatic gas-bearing elements are widely used to achieve high motion accuracy. With appropriate sealing and exhaustion systems, these elements can be used to transfer high-precise movements into a vacuum environment.

Within this paper, a gas bushing with porous flow restrictor is used for the transmission of a mechanical movement into a vacuum chamber. Design and theoretical layout of the necessary sealing and evacuation system are shown. Experimental results prove the feasibility to transmit mechanical movement from normal atmosphere to high vacuum environment (10^{-2} Pa) with two-stage exhaust system.

INTRODUCTION

For positioning processes with an accuracy in the nanometer range, the usage of gas bearing guides is beneficial since friction is negligible and no stick-slip effects occur.

In semiconductor industry applications or for high-precise measurements, some positioning processes have to be performed within a vacuum environment since disturbing influences of gas molecules like absorption effects or the change of the refraction index have to be eliminated.

Different kinds of setups to realize the combination of vacuum and gas bearings are known, e.g. the operation of the whole gas bearing element within the vacuum chamber [1] or the integration of the gas bearing into the vacuum chamber wall [2]. For both operation modes, the flow of gas into the vacuum chamber has to be restricted while keeping the favorable

guiding abilities of the gas bearing. The commonly used method to keep the vacuum quality is to exhaust the supplied gas before reaching the vacuum chamber using a plurality of exhaustion grooves and contact-free seal gaps comprising a high flow resistance [1,2].

The actual investigation realizes the integration of a gas bushing including sealing system and exhaustion pipes into a single device (see FIG.1). The interfaces to supply the pressurized gas and to exhaust the gas from the sealing unit are directly located at the atmospheric side of the system. This setup allows for the usage of standard drives at atmospheric conditions and yet positioning within a high vacuum environment (10^{-2} Pa) with the precision of gas guiding elements.

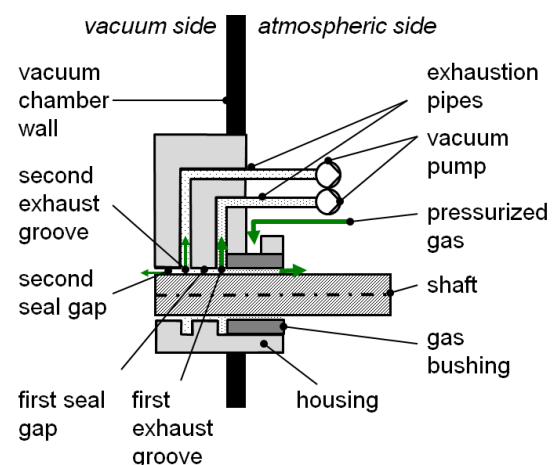


FIGURE 1. Schematic, cross-sectional view of the investigated gas lubricated, mechanical vacuum feedthrough with two-stage exhaust system.

Studies are carried out using an experimental setup, in order to verify the general feasibility of this concept. Analytical calculations enable the comparison of the experimental values with the theoretical pressure distribution. In addition, preliminary results of the motion characteristics are obtained.

ANALYTICAL CALCULATION

In order to make the calculations of the pressure distribution comparable to other systems, the results should be as system independent as possible. The used basic parameters are shown in FIG. 2. At the points of interest (sealing unit with p_1 and p_2 ; vacuum chamber p_c) the pressure is analytically determined. The parameters which have to be known are the mass flow rate $\dot{m}$ of the bushing, the geometrical dimensions of the system as well as the pumping speeds of the connected vacuum pumps. Methods to calculate the pressure in front of the first seal gap using the common relations of flow through pipes are described in various technical books e.g. [3] and some analytical descriptions of the flow through the gap of the sealing systems are brought together by Trost and Heidler [4,5]. The transition between continuous flow to intermediate and molecular flow have to be considered.

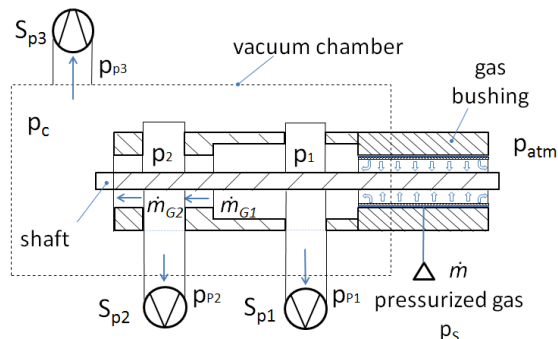


FIGURE 2. Basic parameters for calculation of pressure distribution

The pressure in front of the seal gap depends on the mass flow rate of the bushing. The mass flow rate $\dot{m}$ had been obtained experimentally for different supply pressures¹ p_s and at vacuum ($\dot{m}_{vac}$) and atmospheric ($\dot{m}_{atm}$) conditions to account for the special operating conditions of the bushing (one side vacuum, one side atmosphere). The mean of the mass flow rates

¹ relative to atmosphere (0.1 MPa = 1 bar ~ 14.50 psi)

of the bushing at the according relative pressure during the intended operation was used for the calculations. By way of example, $\dot{m}_{(p_s=0.3 \text{ MPa})} = \dot{m}_{atm(p_s=0.3 \text{ MPa})}/2 + \dot{m}_{vac(p_s=0.4 \text{ MPa})}/2$.

Within the investigated setup, it is assumed that one half of the gas flows into the direction of the vacuum chamber while the other half of the gas escapes to surrounding atmosphere. Hence, $\dot{m}/2$ is used for the analytical calculations. These mass flow rates are compared in TAB.1 to the later measured mass flow rates $\dot{m}_{exp}$ of the actual investigated setup.

TABLE 1. Mass flow rate of the bushing for original operational mode

	$p_{s0.2 \text{ MPa}}$	$p_{s0.3 \text{ MPa}}$	$p_{s0.4 \text{ MPa}}$
$\dot{m} / \text{kg} \cdot \text{s}^{-1}$	2.67e-5	4.67e-5	7.18e-5
$\dot{m}_{exp} / \text{kg} \cdot \text{s}^{-1}$	2.68e-5	4.69e-5	7.09e-5

The mass flow through the seal gap $\dot{m}_G$ is calculated using EQ. (1), approximating the ring-like gap by a gap with rectangular cross section [3]. Hereby A is the cross sectional area, l the length and h the height of the gap, the average velocity of molecules is $\bar{c}$ and the pressure drop along the gap Δp_g . A simplification for this calculation is done by assuming the pressure behind the seal gap to be approximately zero compared to the pressure in front of the seal gap. This procedure was followed for both seal gaps.

$$\dot{m}_G = \frac{2 \cdot A}{\pi \cdot \bar{c}} \frac{1 + \ln(0.433 \cdot l/h + 1)}{l/h + 1} \Delta p_g \quad (1)$$

Finally, the pressure within the vacuum chamber p_c is derived from the mass flow through the last seal gap $\dot{m}_{G2}$ and the pumping speed of the vacuum chamber's pump using the specific gas constant R_s and the temperature T (see EQ.(2)).

$$p_c = \frac{\dot{m}_{G2} \cdot R_s \cdot T}{S_{p3}} \quad (2)$$

SETUP

The whole device shall be attached to the vacuum chamber using the standard DN63 ConFlat flange (ISO 3669). Based on the analytical calculation and on the available space, the design of the experimental setup has to fulfil certain requirements.

A small gap height is favourable to realize a low pressure in the vacuum chamber. The manufacturing and assembly tolerances between the guided shaft and the sealing surface have to be minimized in order to accomplish a small seal gap height. In addition, the whole design and all interfaces have to be vacuum compatible. The CAD-design of the experimental setup is shown in FIG.3 and the picture of the setup in FIG.4.

The employed gas bushing from New Way® has a diameter of $\varnothing 13.022 \text{ mm} \pm 3 \text{ }\mu\text{m}$. The paired steel shaft with a length of 250 mm has a diameter of $\varnothing 12.995 \text{ mm} \pm 5 \text{ }\mu\text{m}$. The bushing is fixed within the housing using an adhesive bonding process. On the vacuum side, the two-stage sealing system is attached. One stage consists of the exhaust groove and the following seal gap (narrow gap with a high flow resistance). By joining the sealing system and the housing with an adjustment bonding process, the precise alignment of the guiding axis of the bushing with the axis of the sealing surfaces is enabled while the gap height is regulated by adjustable spacers. The mean height of the seal gap is $50 \text{ }\mu\text{m}$ and the length of one seal gap is 10 mm.

The exhaust grooves are connected with exhaustion pipe interfaces on the atmospheric side of the housing via interior channels. Each stage of the sealing system is evacuated by a separate vacuum pump. The first exhaust groove is evacuated using a rotary vane pump, the second exhaust groove and the vacuum chamber by using a turbo molecular pump.

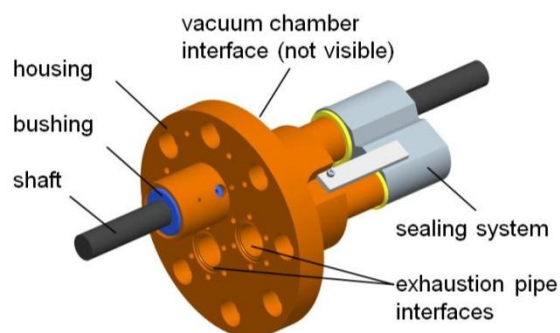


FIGURE 3. CAD-design of vacuum feedthrough using gas bushing and sealing system

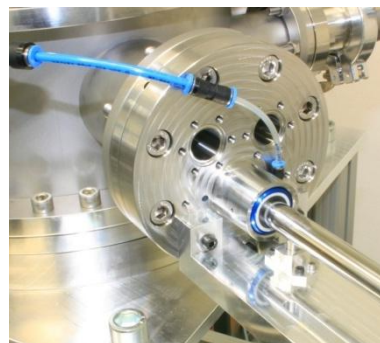


FIGURE 4. Picture of vacuum feedthrough using gas bushing and sealing system

As described in various vacuum technology books (e.g. [3]), the nominal pumping speed S of the vacuum pumps is reduced by the connecting pipes to the effective pumping speed S_{eff} at the point of interest. These pumping speeds are displayed in TAB. 2 for the used setup.

TABLE 2. Nominal pumping speed and effective pumping speed of used vacuum pumps at $p_s = 300000 \text{ Pa}$ (Parameters compare FIG.2)

S_{p1}	Nominal	8.2 l/s
	Effective	6.8 l/s
S_{p2}	Nominal	345 l/s
	Effective	2.2 l/s
S_{p3}	Nominal	67.0 l/s
	Effective	42.4 l/s

Experimental results are gained by feeding the bushing with different supply pressures while evacuating the chamber and the exhaust grooves simultaneously. The pressure is measured using combined Bayard–Alpert / Pirani vacuum gauges. Since these gauges can not be placed directly within the sealing system, the pressure in front of the seal is analytically obtained using the measured pressure in the exhaust pipe. The pressure within the vacuum chamber is recorded after 3 hours of pumping.

RESULTS

Sealing System

The experimental and analytical results of the pressure distribution within the sealing system as well as the pressure in the vacuum chamber are shown in FIG. 5.

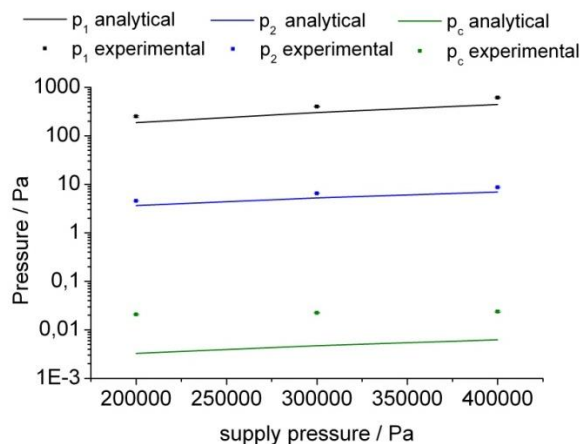


FIGURE 5. Comparison of analytical and experimental pressure for different supply pressures (Parameters compare FIG.2)

It can be outlined, that the analytical and the experimental pressure within the sealing system match each other with a deviation between 19 % and 28%. This is close to the accuracy of the used vacuum gauge ($\pm 25\%$ between 1000 Pa and 1 Pa; 10 % below 1 Pa). The experimental obtained pressure in the vacuum chamber is comparably high since the total pressure had been measured. Thereby, the outgassing of the surfaces of the vacuum chamber has to be taken into account. It can be assumed that after a bake-out procedure and after an elongated evacuation time the pressure within the chamber tends to match the theoretical values within the same range.

Motion Characteristics

The radial displacement for different operation conditions of the shaft in FIG. 6 illustrates the on-going experiments concerning the motion characteristics. Therefore, a capacitive sensor with a resolution of 2 nm was placed close to the bushing at the atmospheric side of the system.

It can be seen that for standard conditions of the bushing (no evacuation of sealing system and chamber) the displacement of the shaft is within the single nanometer range. The supply of the pressurized gas as well as the movement of the shaft does influence the displacement. The peak during the movement $v=1$ ($\triangleq 3 \text{ mm/s}$) indicates the reversal point of the moving direction. Obviously, the major impact is the turning-on of the vacuum pumps. With some decoupling measures, the displacement fell from $\pm 1 \mu\text{m}$ within the range of $\pm 120 \text{ nm}$. Further steps are planned to reduce this influence.

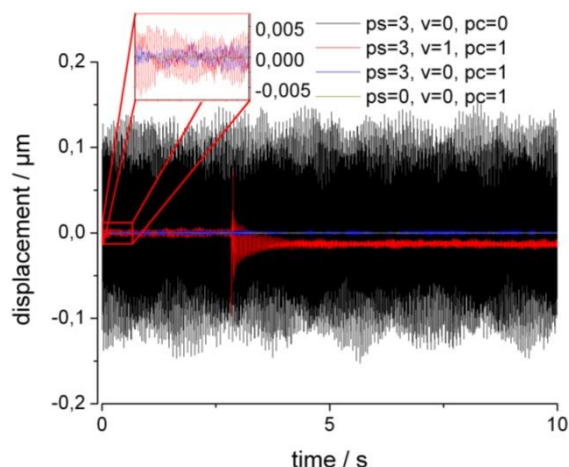


FIGURE 6. Radial displacement of shaft within bushing for different operational conditions ($ps = 0/3$, no supply pressure/ 0.3 MPa supply pressure; $v=0/1$, shaft stands still / shaft moves $pc=0/1$, chamber evacuated / not evacuated)

After the proof of the general feasibility of this concept, a new setup is planned that enables a x-y-positioning of an object in a high vacuum environment. Therefore, additional components for guiding and sealing are required.

ACKNOWLEDGMENT

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MODELING OF THERMAL NUT STIFFNESS IN BALLSCREWS

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ABSTRACT

Thermal nut stiffness model of a ballscrew system is proposed through accurate thermal deformation analysis. Differential temperature distribution is estimated through the finite difference method (FDM). Preload change is computed through interference analysis between shaft and nut sections. Thermal nut stiffness is estimated according to assembly, load and heat transfer conditions.

INTRODUCTION

Positioning accuracy of the ballscrew system depends upon axial stiffness of the feed drive. To compensate for the positioning error, and to predict accuracy of precision machines during the design process, accurate stiffness estimation of the feed drive equipped with ballscrews is required. When they are in contact, the presence of deformation, friction, and lubrication condition generates heat. To calibrate its thermal deformation, temperature estimation of the ballscrew is required. Thermal resistance between balls and grooves, and the change of heat generation of the ballscrew mechanism should be estimated according to the assembly and operating conditions. In addition, radial and axial deformations of the ballscrew change thermal stiffness, and then the loading condition and the nut stiffness are varied. These effects change the nut stiffness model [1-4].

In this paper, the heat generation mechanism, temperature analysis module, and thermal deformation module are accurately derived through the analysis of heat generation and transfer mechanism. A complete ballscrew nut stiffness model including thermal deformation effect is proposed according to assembly, loading and operating conditions. Differential temperature distribution is estimated through the finite difference model [2-3] and proper boundary conditions obtained from the drag torque and operating conditions. Preload change due to the differential growth between the nut

and shaft is obtained from the differential temperature. Applying the preload change to the previously developed nut stiffness model [1], thermal stiffness is estimated. Several case studies are performed to verify the validity of the proposed method.

THERMAL MODELING

Figure 1 shows a double-nut ballscrew feed drive system. Figure 2 shows contact states of the i^{th} -ball and groove. α represents the contact angle between the ball and the groove, F_t is the axial thrust force, and F_i denotes contact ball load. These geometric parameters are used in calculation of Hertzian deformation. As the ballscrew rotate, heat is generated due to slip between balls and grooves, viscous friction due to shear of lubricant, rolling contact friction. In addition, support bearings generates frictional heat. The nut stiffness is varied according to these frictions.

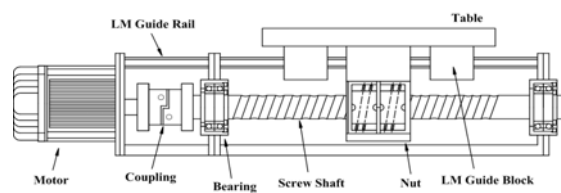


FIGURE 1. Double-nut Ballscrew Feed Drive.

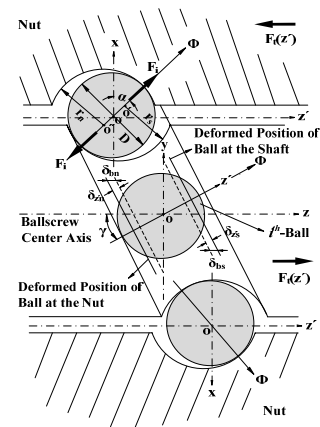


FIGURE 2. Geometry and Contact Mechanism.

Finite Difference Method (FDM)

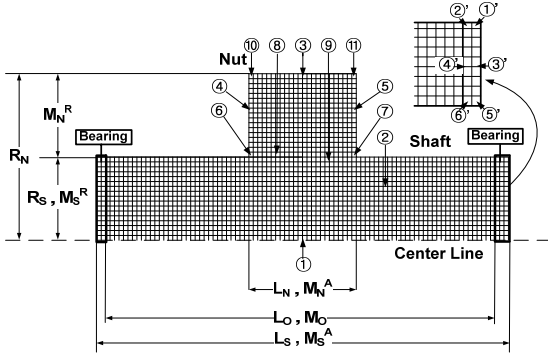


FIGURE 3. Axisymmetric FDM Model of the Ballscrew.

Comparing with the previous model [3], Fig. 3 shows more detail FDM model with smaller axial and radial difference Δx and Δr , respectively. Table 1 describes parameters of the FDM model. Considering heat equilibrium conditions, heat conduction in a solid, heat convection around a surface, and frictional heat generation from the support bearings and contact surfaces between balls and grooves, explicit form of finite difference equations are given by Eq. (1). $T_{m,n}^F$ and $T_{m,n}$ are next and present step temperatures, respectively. The friction torque and temperatures at the contact part of the bearing and the screw are given by part 1', 2', 6, 7, 8 and 9 equations of Eq. (1). Eq. (2) should be satisfied to get temperature distribution by FDM. Fig. 4 shows temperature distribution after one hour operation. Fig. 5 shows contact points temperature record according to preload, rotational speed and operation time. Temperature reaches steady state around 60 minutes. It is confirmed that temperature depends upon not only assembly and load conditions but also heat convection and conduction conditions in the ballscrew assembly.

$$\begin{aligned}
 1: T_{m,n}^F &= T_{m,n} + Fo_x(T_{m,n+1} + T_{m,n-1} - 2T_{m,n}) + 4Fo_r(T_{m,n-1} - T_{m,n}) \\
 2: T_{m,n}^F &= T_{m,n} + Fo_x(T_{m,n+1} + T_{m,n-1} - 2T_{m,n}) \\
 &\quad + Fo_r(k_1T_{m+1,n} + k_2T_{m-1,n} - 2T_{m,n}) \\
 3: T_{m,n}^F &= T_{m,n} + Fo_x(T_{m,n+1} + T_{m,n-1} - 2T_{m,n}) + Fo_rk_3(T_{m+1,n} - T_{m,n}) \\
 &\quad + Fo_r \cdot Bi_r \cdot k_4(T_{\infty} - T_{m,n}) \\
 4: T_{m,n}^F &= T_{m,n} + Fo_x(k_1T_{m+1,n} + k_2T_{m-1,n} - 2T_{m,n}) + Fo_x(T_{m,n+1} - T_{m,n}) \\
 &\quad + Fo_x \cdot Bi_x(T_{\infty} - T_{m,n}) \\
 5: T_{m,n}^F &= T_{m,n} + Fo_x(k_1T_{m+1,n} + k_2T_{m-1,n} - 2T_{m,n}) + Fo_x(T_{m,n-1} - T_{m,n}) \\
 &\quad + Fo_x \cdot Bi_x(T_{\infty} - T_{m,n})
 \end{aligned}$$

$$\begin{aligned}
 6: T_{m,n}^F &= T_{m,n} + Fo_x(T_{m,n+1} - T_{m,n}) + Fo_rk_5(T_{m-1,n} - T_{m,n}) \\
 &\quad + Fo_x \cdot Bi_x(T_{\infty} - T_{m,n}) + R \frac{T_{m,n-1}V_s - T_{m,n}V_n}{V_n} + T_{fn}
 \end{aligned} \quad (1)$$

$$\begin{aligned}
 7: T_{m,n}^F &= T_{m,n} + Fo_x(T_{m,n-1} - T_{m,n}) + Fo_rk_5(T_{m-1,n} - T_{m,n}) \\
 &\quad + Fo_x \cdot Bi_x(T_{\infty} - T_{m,n}) + R \frac{T_{m,n-1}V_s - T_{m,n}V_n}{V_n} + T_{fn}
 \end{aligned}$$

$$\begin{aligned}
 8: T_{m,n}^F &= T_{m,n} + Fo_x(T_{m,n+1} + T_{m,n-1} - 2T_{m,n}) + Fo_rk_5(T_{m+1,n} - T_{m,n}) \\
 &\quad + R \frac{T_{m+1,n}V_s - T_{m,n}V_n}{V_n} + T_{fn}
 \end{aligned}$$

$$\begin{aligned}
 9: T_{m,n}^F &= T_{m,n} + Fo_x(T_{m,n+1} + T_{m,n-1} - 2T_{m,n}) + Fo_rk_3(T_{m+1,n} - T_{m,n}) \\
 &\quad + R \frac{T_{m-1,n}V_n - T_{m,n}V_s}{V_s} + T_{fs}
 \end{aligned}$$

$$\begin{aligned}
 10: T_{m,n}^F &= T_{m,n} + Fo_x(T_{m,n+1} - T_{m,n}) + Fo_rk_3(T_{m+1,n} - T_{m,n}) \\
 &\quad + (Fo_x \cdot Bi_x + k_4 \cdot Fo_r \cdot Bi_r)(T_{\infty} - T_{m,n})
 \end{aligned}$$

$$\begin{aligned}
 11: T_{m,n}^F &= T_{m,n} + Fo_x(T_{m,n-1} - T_{m,n}) + Fo_rk_3(T_{m+1,n} - T_{m,n}) \\
 &\quad + (Fo_x \cdot Bi_x + k_4 \cdot Fo_r \cdot Bi_r)(T_{\infty} - T_{m,n})
 \end{aligned}$$

$$\begin{aligned}
 1': T_{m,n}^F &= T_{m,n} + Fo_x(T_{m,n-1} - T_{m,n}) + Fo_rk_3(T_{m+1,n} - T_{m,n}) \\
 &\quad + (Fo_x \cdot Bi_x + k_4 \cdot Fo_r \cdot Bi_r)(T_{\infty} - T_{m,n}) + T_{fb}
 \end{aligned}$$

$$\begin{aligned}
 2': T_{m,n}^F &= T_{m,n} + Fo_x(T_{m,n+1} + T_{m,n-1} - 2T_{m,n}) + Fo_rk_3(T_{m+1,n} - T_{m,n}) \\
 &\quad + Fo_r \cdot Bi_r \cdot k_4(T_{\infty} - T_{m,n}) + T_{fb}
 \end{aligned}$$

$$\begin{aligned}
 3': T_{m,n}^F &= T_{m,n} + Fo_x(k_1T_{m+1,n} + k_2T_{m-1,n} - 2T_{m,n}) + Fo_x(T_{m,n-1} - T_{m,n}) \\
 &\quad + Fo_x \cdot Bi_x(T_{\infty} - T_{m,n})
 \end{aligned}$$

$$4': T_{m,n} + Fo_x(T_{m,n+1} + T_{m,n-1} - 2T_{m,n}) + Fo_r(k_1T_{m,n-1} + k_2T_{m+1,n} - 2T_{m,n})$$

$$5': T_{m,n}^F = T_{m,n} + Fo_x(T_{m,n-1} - 2T_{m,n}) + 4Fo_r(T_{m-1,n} - T_{m,n})$$

$$6': T_{m,n}^F = T_{m,n} + Fo_x(T_{m,n+1} + T_{m,n-1} - 2T_{m,n}) + 4Fo_r(T_{m-1,n} - T_{m,n})$$

where,

$$k_1 = 1 - \frac{0.5}{n}, \quad k_2 = 1 + \frac{0.5}{n}, \quad k_3 = \frac{4(2n-1)}{4n-1}, \quad k_4 = \frac{8n}{4n-1}, \quad k_5 = \frac{4(2n+1)}{4n+1}$$

$$Fo_x = \frac{\beta \Delta t}{(\Delta x)^2}, \quad Fo_r = \frac{\beta \Delta t}{(\Delta r)^2}, \quad Bi_x = \frac{h \Delta x}{k}, \quad Bi_r = \frac{h \Delta r}{k}, \quad \beta = \frac{\kappa}{\rho c}$$

$$V_{s,b} = \frac{\pi(n-0.25)(\Delta r)^2 \Delta x}{2}, \quad V_n = \frac{\pi(n+0.25)(\Delta r)^2 \Delta x}{2}$$

$$H_f = 1.4047 \times 10^{-4} \cdot n \cdot M_t, \quad T_{fs} = \frac{H_f \Delta t}{c \rho V_s}, \quad T_{fn} = \frac{H_f \Delta t}{c \rho V_n}$$

$$1 \Rightarrow 1 - 2Fo_x - 4Fo_r \geq 0$$

$$2 \Rightarrow 1 - 2Fo_x - 2Fo_r \geq 0$$

$$3 \Rightarrow 1 - 2Fo_x - Fo_r \cdot k_3 - Fo_x \cdot Bi_r \cdot k_4 \geq 0$$

$$4 = 5 \Rightarrow 1 - 2Fo_r - Fo_x - Fo_x \cdot Bi_r \geq 0$$

$$6 = 7 \Rightarrow 1 - Fo_x - Fo_r \cdot k_5 - Fo_x \cdot Bi_r - R \geq 0$$

$$8 \Rightarrow 1 - 2Fo_x - Fo_r \cdot k_5 - R \geq 0$$

$$9 \Rightarrow 1 - 2Fo_x - Fo_r \cdot k_3 - R \geq 0$$

$$10 = 11 \Rightarrow 1 - Fo_x - Fo_r \cdot k_3 - Fo_x \cdot Bi_r - Fo_x \cdot Bi_r \cdot k_4 \geq 0$$

$$1' \Rightarrow 1 - Fo_x - Fo_r \cdot k_3 - Fo_x \cdot Bi_r - Fo_x \cdot Bi_r \cdot k_4 \geq 0$$

$$2' \Rightarrow 1 - 2Fo_x - Fo_r \cdot k_3 - Fo_x \cdot Bi_r \cdot k_4 \geq 0$$

$$3' \Rightarrow 1 - 2Fo_r - Fo_x - Fo_x \cdot Bi_r \geq 0$$

$$4' \Rightarrow 1 - 2Fo_x - 2Fo_r \geq 0$$

$$5' \Rightarrow 1 - Fo_x - Fo_r \cdot k_5 - Fo_x \cdot Bi_r - R \geq 0$$

$$6' \Rightarrow 1 - 2Fo_x - 4Fo_r \geq 0$$

(2)

TABLE 1. Parameters of the FDM Model.

	Model 1	Model 2
Shaft Length, L_S [mm]	1000	
Lead [mm]	5	10
Total Ball Number, Z	240	90
Nut Length, L_N [mm]	105	200
Shaft Outer Diameter, R_S [mm]	25	36
Nut Outer Diameter, R_N [mm]	50	75
Radial Difference, Δr [mm]	0.5	
Axial Difference, Δx [mm]	2.5	
Number of Shaft Mesh in Radial Direction, M_S^R	25	36
Number of Nut Mesh in Radial Direction, M_N^R	25	39
Number of Shaft Mesh in Axial Direction, M_S^A	400	
Number of Nut Mesh in Axial Direction, M_N^A	42	80
Operation Length, L_O [mm]	988	
Preload, F_p [Kgf]	70 , 129	
Contact Angle, α° [°]	45	
Lead Angle, γ° [°]	5	
Kinematic Viscosity of Lubricant, ν_0 [cSt]	32	
Ball Screw Rotational Speed, n [rpm]	250~2500	
Frictional Coefficient, μ	0.003	
Ball Load, Q	F_a / Z	
Specific Heat, c [J/Kg°C]	460	
Thermal Conductivity, κ [W/m°C]	60.5	
Density, ρ [Kg/m³]	7865	
Thermal Expansion Coefficient, α [°C⁻¹]	1.2×10^{-6}	

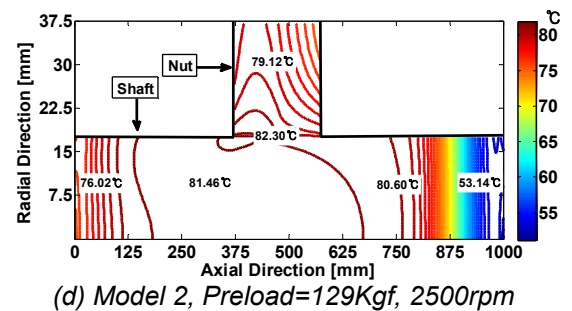
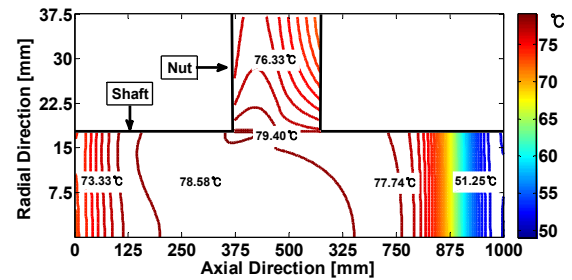
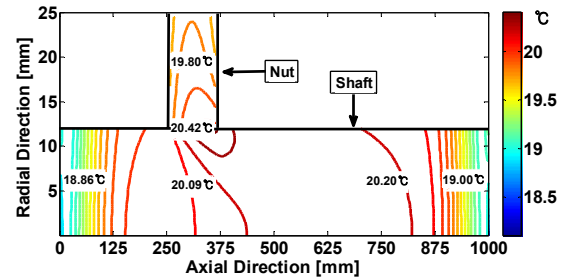
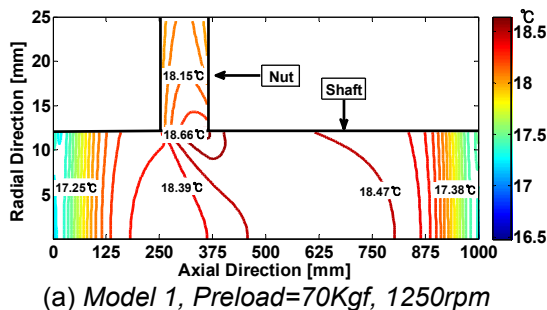


FIGURE 4. Temperature Distributions of the Ballscrew Assembly after 1 Hour Operation.

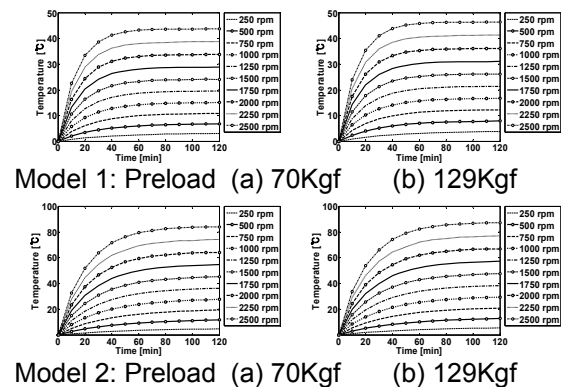


FIGURE 5. Contact Points Temperature Record.

Heat Generation Mechanism

Total frictional torque caused by applied load, viscosity and spinning motion is given by Eq. (3). Tables 3 and 4 show frictional and convection coefficients z , y and f_0 given in Eq. (3) [4-5].

$$\begin{aligned}
F_{\beta} &= 0.9 \cdot F_a \cdot \cot(\alpha) \\
f_a &= z \cdot (F_s / C_s)^y \\
F_s &= Y_s \cdot F_a \\
M_a &= f_a \cdot F_{\beta} \cdot d_m \\
M_v &= 10^{-7} \cdot f_0 \cdot (v_0 n)^{2/3} \cdot d_m^3 & v_0 \cdot n \geq 2000 \\
M_v &= 160 \times 10^{-7} \cdot f_0 \cdot d_m^3 & v_0 \cdot n \leq 2000 \\
M_s &= \frac{3\mu Q a \varepsilon}{8}
\end{aligned} \tag{3}$$

TABLE 3. Values of z and y .

Bearing Type	Contact Angle	z	y
Angular Contact	30-40°	0.001	0.33

TABLE 4. f_0 in Different Lubrication Conditions.

Angular Contact Ball Bearing	Grease	Oil Mist	Oil Bath	Oil Bath or Oil Jet
	2	1.7	3.3	6.6

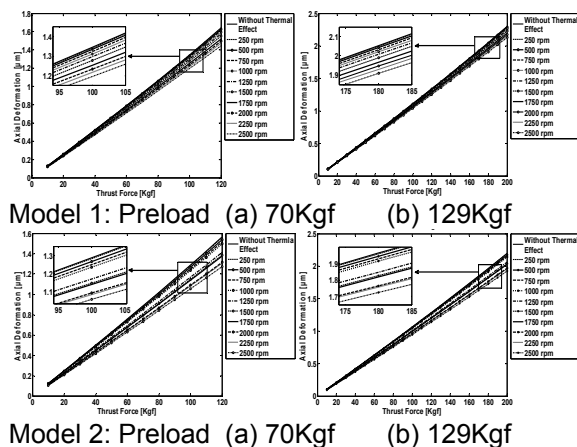


FIGURE 6. Axial Deformation Changes.

THERMAL NUT STIFFNESS

According to thermal analysis results shown in Fig. 4, temperature of shaft section is higher than nut section. This means thermal expansion of the shaft section is greater than the nut section in radial and axial directions. Hertzian deformation occurred in the ballscrew is changed owing to this interference. Axial deformation of the ballscrew varies according to Hertzian deformation changes as shown in Fig. 6. Differential preload due to the interference is generated between the shaft and the nut. Nut stiffness is computed with respect to preload changes through the previous research [1]. Fig. 7 shows thermal nut stiffness according to assembly, load and operating conditions. It confirms that thermal nut stiffness increases as

the interference increases according to preload and rotational speed increases.

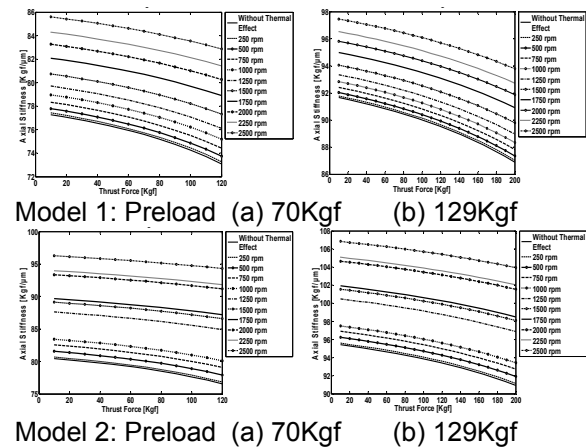


FIGURE 7. Thermal Nut Stiffnesses.

CONCLUSIONS

Thermal deformation analysis of a ballscrew system has been conducted. Interference between the screw and nut sections generates preload change. This varies thermal nut stiffness. Validity of the thermal nut stiffness model is verified through several kinds of simulation.

ACKNOWLEDGEMENT

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MODELING AND UNCERTAINTY ESTIMATION OF BALLSCREW NUT STIFFNESS

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ABSTRACT

Uncertainty estimation of the nut stiffness of a ballscrew is conducted through Monte Carlo method. Expanded uncertainty of the stiffness model is estimated by stochastic analysis of uncertainty effects. Experimental verification has been conducted through GUM.

INTRODUCTION

Positioning accuracy of a ballscrew system is subject to nut stiffness. It depends upon Hertzian deformation of balls and grooves, and body deformation of screw shaft and nut according to thrust force, preload and groove characteristics [1]. Fig. 1 shows contact state of the i^{th} -ball and grooves. α represents the contact angle of the ball and groove, D is the ball diameter. F_t is the axial thrust force, F_i denotes ball contact load, etc. Using the given parameters, Hertzian contact deformation constants between balls and grooves are derived through elliptic integrations [1-2].

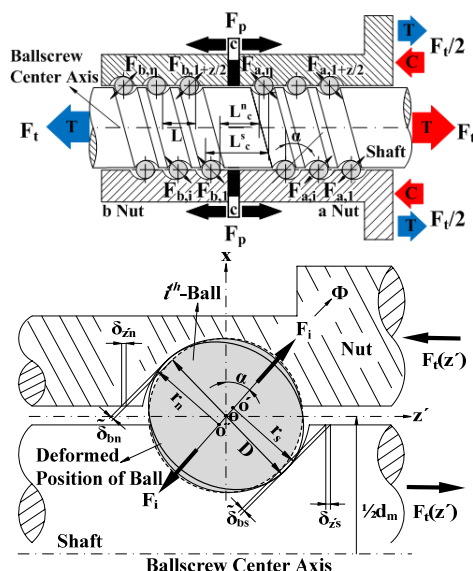


FIGURE 1. Double-nut Ballscrew and Contact Mechanism.

Applying for compatibility and force equilibrium conditions, body deformation is estimated according to the assembly and load conditions as shown in Fig. 2 [1]. However, variation of the conformity affects contact deformation significantly. Fluctuation of the contact angle generates uncertainty of contact deformation estimates [1-2]. In this paper, accurate modeling and uncertainty estimation of the ballscrew nut stiffness model is conducted through Monte Carlo method (MCM).

UNCERTAINTY ESTIMATION

To account for uncertainty of various conditions, calculation method of nut stiffness related to geometric uncertainty factors should be proposed. In addition to previous research analyzing the nut stiffness of ballscrew [1-2], this paper proposes Monte Carlo simulation to estimate the uncertainty as shown in Fig. 3.

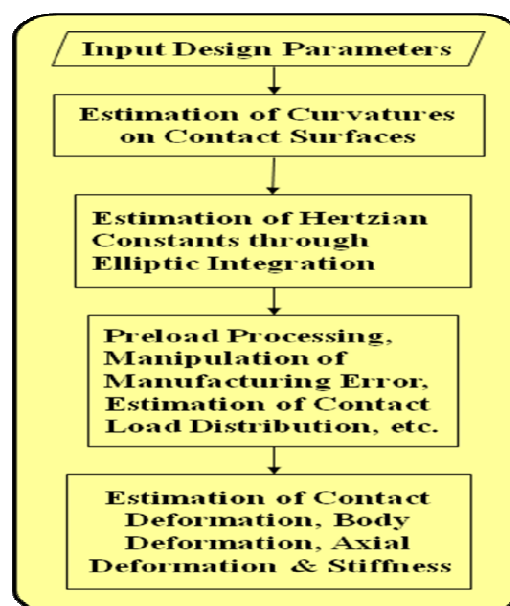


FIGURE 2. Computational Procedure of Ballscrew Nut Stiffness.

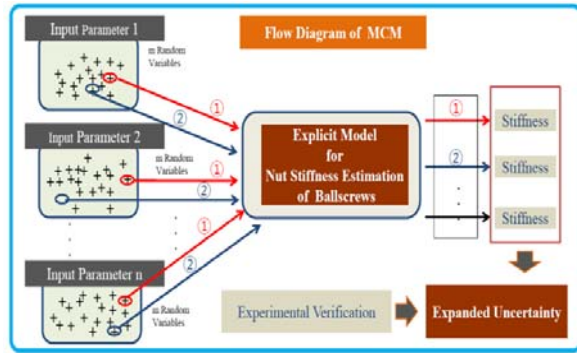


FIGURE 3. Uncertainty Estimation Process.

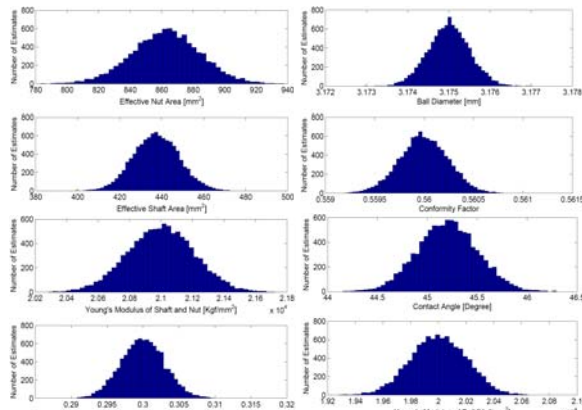


FIGURE 4. Uncertainty Factors for Stiffness Estimation through MCM.

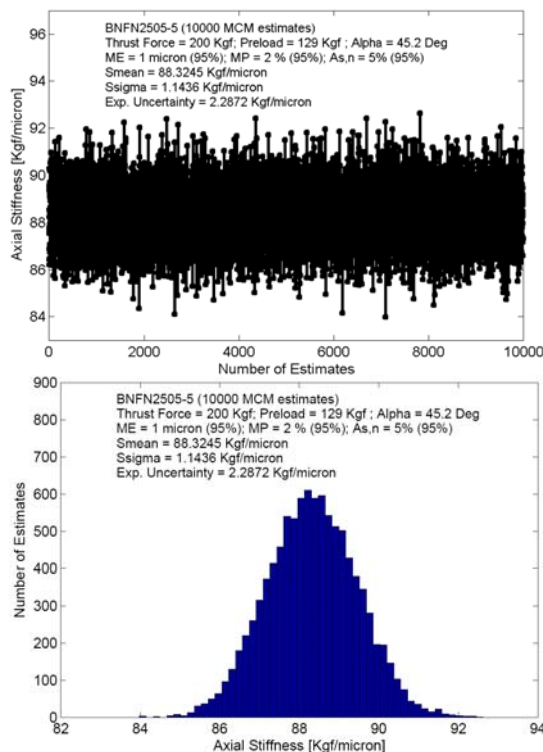


FIGURE 5. Estimated Uncertainty of Stiffness through MCM.

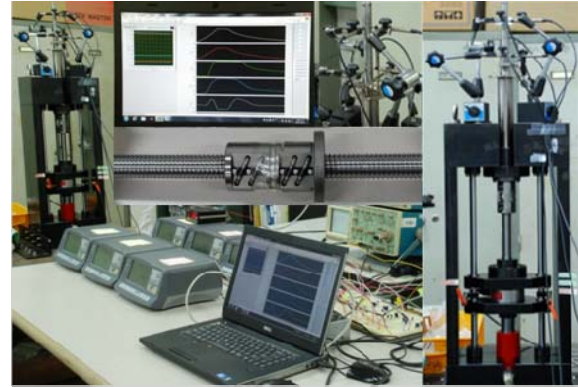


FIGURE 6. Measurement System of Ballscrew Nut Stiffness.

THK's double-nut BNFN2505-5 with tensile preload 129Kgf and CC assembly condition is selected for stiffness estimation. Uncertainty factors such as contact angle α , ball diameter d , conformity factor $f_{s,n}$, effective area $A_{s,n}^{eff}$ of the screw shaft and the nut, elastic modulus, etc. are selected for MCM stiffness estimation. Fig. 4 shows histograms of uncertainty factors as input parameters. Fig. 5 shows 10,000 times of MCM uncertainty estimation results of the nut stiffness for 200Kgf thrust force. Mean stiffness of 90.87 and 88.32 Kgf/ μm is obtained for thrust force of 100 and 200 Kgf, respectively. Expanded uncertainty for 200 Kgf thrust force is obtained between 86.03 and 90.61 Kgf/ μm with 95.46% confidence.

EXPERIMENTAL SETUP

To measure ballscrew stiffness accurately, experiments have been conducted on a measurement system composed of a hydraulic load unit, a load cell, 5 displacement sensors (Tesa, GT21HP) and a data acquisition system (NI, Labview) as shown in Fig. 6. Fig. 7 shows tilting error and tilting angle measurement system. During experiments, titling of the ballscrew axis occurs between the nut and the screw shaft owing to loading and clearance between the nut and ballscrew shaft. Measurement results are changed from true deformations to false deformation amounts because of the sensor target movement shown in Fig. 7. To correct the measurement deformations to true values, identification of tilting angles θ and ϕ of the ballscrew with respect to x and y axis is required first. Trigonometric processing of measurement data obtained from B, C, D and E displacement sensors should be performed. After calibration of

the tilting error, compensation of the axial deformation of the ballscrew is conducted according to the following equation:

$${}^{CS1}P_{CS2} = [R_Y] \cdot [R_X] \cdot {}^{CS1}P_{CS2}$$

$$\begin{bmatrix} L+\Delta x \\ 0+\Delta y \\ L_T+\Delta z \\ 1 \end{bmatrix} = \begin{bmatrix} \cos\varphi & \sin\varphi\sin\theta & \sin\varphi\cos\theta & 0 \\ 0 & \cos\theta & -\sin\theta & 0 \\ -\sin\varphi & \cos\varphi\sin\theta & \cos\varphi\cos\theta & 0 \\ 0 & 0 & 0 & 1 \end{bmatrix} \begin{bmatrix} L \\ 0 \\ L_T \\ 1 \end{bmatrix} \quad (1)$$

In Eq. (1), ΔZ is the true axial deformation of the ballscrew. Fig. 8 shows 30times axial deformations and stiffnesses with and without tilting errors.

EXPERIMENTAL VERIFICATION BY GUM

Experimental results are compared with MCM estimated stiffness by using GUM. Fig. 9 shows GUM procedure [3-4]. There are many uncertainty factors classified as A and B types. A type is uncertainty contribution during measurement of thrust force and axial deformation as given by Eq. (2). B type is uncertainty contribution by sensor resolution and sensor uncertainty. Setup error of sensors and thermal expansion by environment as shown in Fig. 7 are given by Eq. (3).

$$u_{c1} = C_{\text{Deformation}} u_{\text{Deformation}} = \frac{\partial K}{\partial \delta_i} s(\bar{\delta})$$

$$u_{c2} = C_{\text{Thrust Force}} u_{\text{Thrust Force}} = \frac{\partial K}{\partial F_i} s(\bar{F}) \quad (2)$$

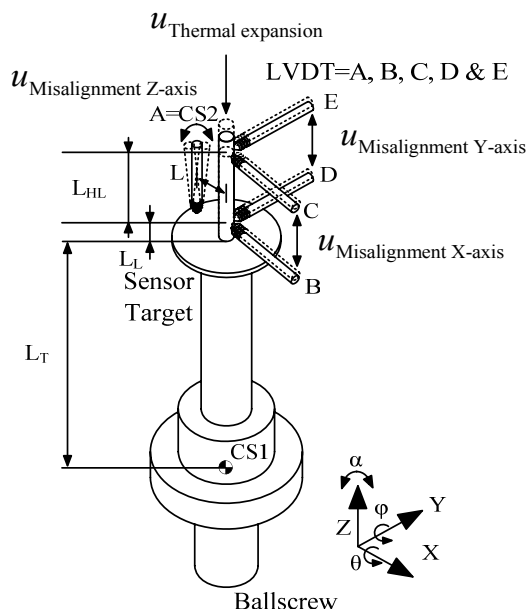


FIGURE 7. 5 Sensors for Tilting Measurement.

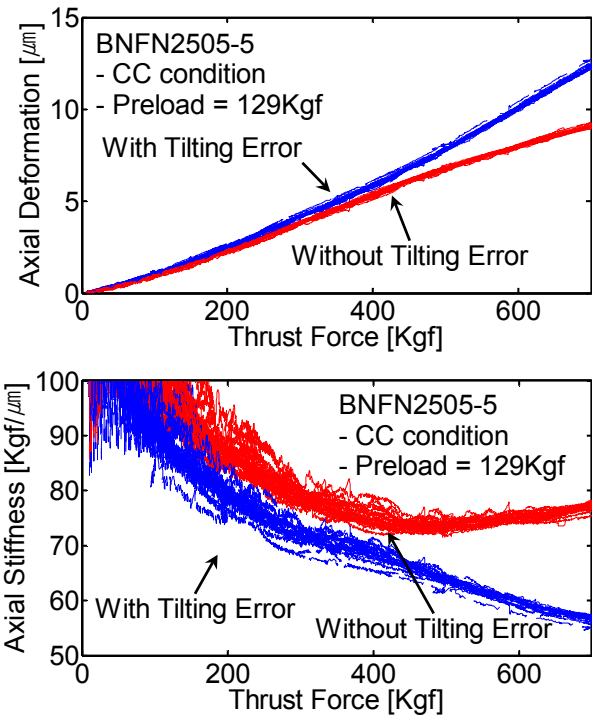


FIGURE 8. Axial Deformations and Stiffnesses Before and After Compensation.

$s(\bar{\delta})$ and $s(\bar{F})$ are standard uncertainties of the deformation and thrust force. Sensitivities of B type are obtained by measurements. Standard uncertainties of B type given by Eq. (3) are 0.29, 0.02, 1.15, 0.33, 0.33 and 0.06, respectively. Combined uncertainty is given by Eq. (4).

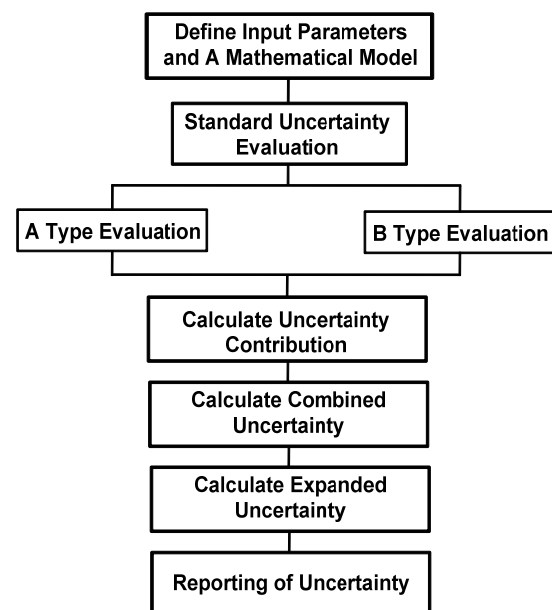


FIGURE 9. Uncertainty Evaluation Steps by GUM.

$$\begin{aligned}
u_{c3} &= c_{\text{Loadcell}} u_{\text{Loadcell}} = \frac{\partial K}{\partial F} \frac{R_{\text{Loadcell}}}{\sqrt{3}} / 2 \\
u_{c4} &= c_{\text{LVDT}} u_{\text{LVDT}} = \frac{\partial K}{\partial \delta} \frac{U_{\text{LVDT Calibration}}}{k_{\text{LVDT Calibration}}} \\
u_{c5} &= c_{\text{Misalignment Z-axis}} u_{\text{Misalignment Z-axis}} = \frac{\partial K}{\partial \alpha} \frac{R_{\text{Misalignment Z-axis}}}{\sqrt{3}} / 2 \\
u_{c6} &= c_{\text{Misalignment X-axis}} u_{\text{Misalignment X-axis}} = \frac{\partial K}{\partial \phi} \frac{R_{\text{Misalignment X-axis}}}{\sqrt{3}} / 2 \\
u_{c7} &= c_{\text{Misalignment Y-axis}} u_{\text{Misalignment Y-axis}} = \frac{\partial K}{\partial \theta} \frac{R_{\text{Misalignment Y-axis}}}{\sqrt{3}} / 2 \\
u_{c8} &= c_{\text{Thermal Expansion}} u_{\text{Thermal Expansion}} = \frac{\partial K}{\partial T} \frac{R_{\text{Thermal Expansion}}}{\sqrt{3}} / 2
\end{aligned} \quad (3)$$

$$\begin{aligned}
u_c^2 &= c_{\text{Thrust Force}}^2 u_{\text{Thrust Force}}^2 + c_{\text{Deformation}}^2 u_{\text{Deformation}}^2 \\
&+ c_{\text{Loadcell}}^2 u_{\text{Loadcell}}^2 + c_{\text{LVDT}}^2 u_{\text{LVDT}}^2 \\
&+ c_{\text{Misalignment Z-axis}}^2 u_{\text{Misalignment Z-axis}}^2 \\
&+ c_{\text{Misalignment X-axis}}^2 u_{\text{Misalignment X-axis}}^2 \\
&+ c_{\text{Misalignment Y-axis}}^2 u_{\text{Misalignment Y-axis}}^2 \\
&+ c_{\text{Thermal expansion}}^2 u_{\text{Thermal expansion}}^2
\end{aligned} \quad (4)$$

Expanded uncertainty and confidence level of stiffness S_K are given by

$$\begin{aligned}
U &= k u_c \\
S_K - U &\leq S_K \leq S_K + U
\end{aligned} \quad (5)$$

TABLE 1. Stiffness Uncertainty Measurements according to Thrust Force by GUM.

Thrust Force [Kgf]	Combined Uncertainty u_c [Kgf/ μ m]	Expanded Uncertainty U [Kgf/ μ m]	Confidence Interval [Kgf/ μ m]
100	6.84	13.68	79.2≤92.9≤106.6
200	3.07	6.14	83.2≤89.3≤95.5
300	1.92	3.85	83≤86.9≤90.7
400	1.23	2.45	77.9≤80.4≤82.8
500	0.96	1.91	74.9≤76.8≤78.7
600	0.74	1.48	74.4≤75.8≤77.3
700	0.78	1.55	73.7≤75.3≤76.8

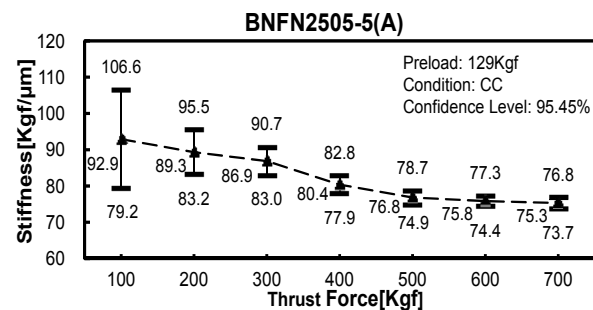


FIGURE 10. Experimental Stiffness obtained by GUM.

As the effective degree of freedom is 30, k is selected 2 and confidence level is 95.45%. Table 1 and Fig. 10 show stiffness results obtained by GUM. Stiffness uncertainty estimations by MCM are included in the confidence interval of experimental uncertainty results obtained by GUM. It is confirmed that improved stiffness measurement setup [2] and tilting compensation method generate good stiffness measurement results. It has been confirmed that uncertainty estimation through MCM proposed in this paper gives correct stiffness results.

CONCLUSIONS

To measure the ballscrew nut stiffness accurately and reduce uncertainty factors in the measurement, improved stiffness measurement setup using 5 displacement sensors and the tilting error compensation algorithm have been devised in this paper. Uncertainty estimation from the previously developed theoretical stiffness model through MCM gives correct stiffness estimation results comparing with the experimental GUM results. It is confirmed that previously developed mathematical stiffness model of ballscrews is correct.

ACKNOWLEDGEMENT

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LOW COST HYDROSTATIC BEARING MANUFACTURING

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ABSTRACT

Manufacturing hydrostatic bearings often requires precision processes that increase the cost of the bearings over other bearing options. Previously, work was completed on a half cylinder bearing made from cutting patterns into a sheet of ultra high molecular weight polyethylene (UHMW) [1]. Initial attempts involved machining the inner surface of an extruded 6061 T651 aluminum tube. The tube was then cut in half axially and the UHMW with the bearing features was adhered to the inner surface. These attempts failed due to residual stresses in the tube distorting the part after it was cut in half. These failures led to a thermoforming technique where the UHMW was adhered to a housing CNC machined out of a solid billet. A master shaft was then placed on the UHMW and the entire assembly was heated. The shaft weight and heat formed the plastic to the master shaft. This paper further investigates replication methods for the manufacturing of the half cylinder hydrostatic bearings.

EPOXY FILLING

Another way to overcome the errors in the shape of the housing was pursued by filling behind the plastic with a thin layer of epoxy resin compound. The epoxy fills in any form errors in the housing while the plastic is held against a master shaft. For testing instead of UHMW, Turcite B¹ was used for the bearing surface. Turcite B is a PTFE based polymer designed for bearings and ways. It has good machinability and wear resistance and is less stiff than UHMW. The Turcite B was tensioned around a master shaft with a mesh screen. The housing was made of an extruded aluminum tube which was cut in half axially with no machining done on the inner diameter. The split tube housing was bonded to the Turcite B through the mesh screen with Waylock II epoxy which is recommended for Turcite B applications. Figure 1 shows the setup for this manufacturing method. No heat is applied to the plastic to avoid any issues with thermal gradients or overheating. A 180° partial arc plain journal bearing was made

and measured on a Zeiss Eclipse CMM. The resulting surface had a cylindricity of 13 μm and was within 2 μm of the master shaft in diameter.

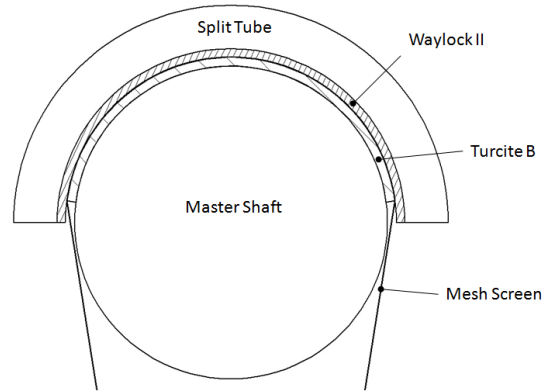


FIGURE 1. Bearing manufacturing method with epoxy (figure not to scale)

COMPLIANT HOUSING

An adjustable housing was also pursued. This design is purposely compliant to allow small changes in the bearing diameter. A sketch of the concept is shown in Figure 2. There is a set of inner leaves, labeled 1, and outer leaves, labeled 2. A screw system is used to spread the leaves apart or draw them together as needed. The screws can be adjusted to allow small deformations of the bearing diameter until the desired shape is achieved. To manufacture this housing, the shape can be cut from flat stock material with a water jet. For longer cylinders, pieces can be stacked and held together with long bolts.

THEORY

For an order of magnitude estimate of the diameter change, the leaves were modeled as a curved cantilever beam as shown in Figure 3. The force from the adjustment screw is represented by the point load, Q . From the sectioned beam shown in Figure 4, the internal moment, M_R , can be solved for.

$$M_R = QR \sin \theta \quad (1)$$

¹www.tss.trelleborg.com

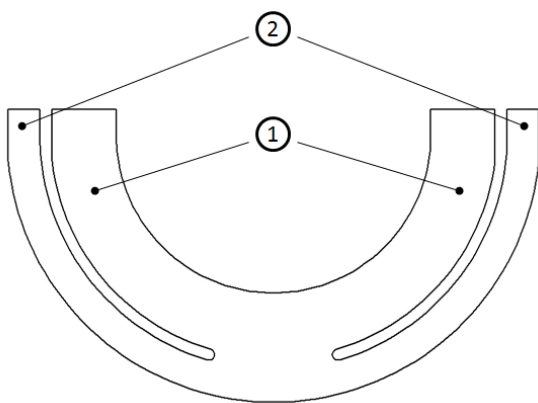


FIGURE 2. Concept sketch of the compliant housing

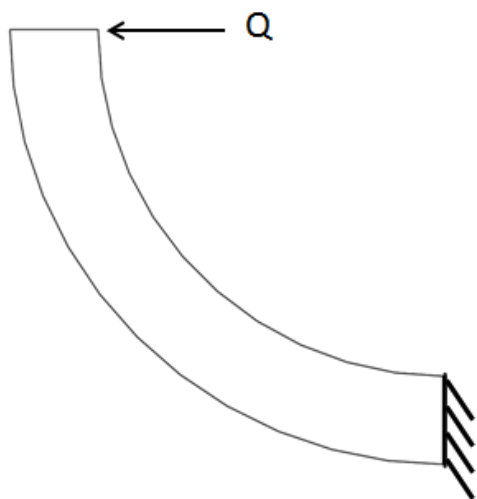


FIGURE 3. Cantilever beam model of housing

The bending energy can then be found.

$$\begin{aligned}
 U_{bending} &= \frac{1}{2EI} \int_0^{\pi/2} M_R^2 R d\theta \\
 &= \frac{1}{2EI} \int_0^{\pi/2} (QR \sin\theta)^2 R d\theta \\
 &= \frac{\pi Q^2 R^3}{8EI} \\
 &= \frac{3\pi Q^2 R^3}{2Ebt^3} \quad (2)
 \end{aligned}$$

The deflection, δ , in the direction of the applied load, Q , is found by taking the partial derivative of the bending energy with respect to Q according

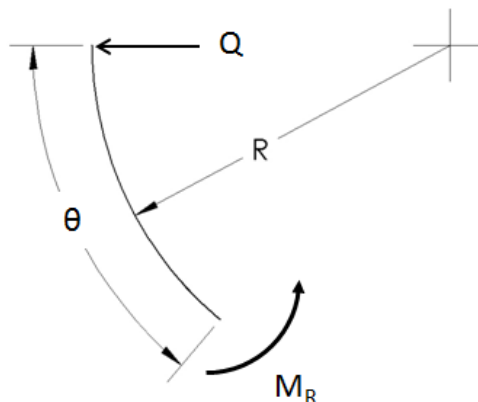


FIGURE 4. Freebody diagram of a section of the beam

to Castigliano's Theorem.

$$\begin{aligned}
 \delta &= \frac{\partial U}{\partial Q} \\
 &= \frac{3\pi R^3}{Ebt^3} Q \quad (3)
 \end{aligned}$$

From (3) a transmission ratio for the two leaves of the compliant housing can be calculated. The modulus, E , and width into the page, b , are considered constant for the stock material. The force, Q , acting on each leaf is the same since it is the reaction that acts on the other leaf. The transmission ratio can be formulated as follows.

$$\frac{\delta_2}{\delta_1} = \left(\frac{t_1 R_2}{t_2 R_1} \right)^3 \quad (4)$$

It should be noted here that the deflection is only a rough estimate of the change in diameter of the housing. It is assumed that the force, Q , is in the direction of the diameter, so that the deflection is the change in the diameter. However, the beam will not deflect with constant curvature, so the cylindricity of the surface will get worse.

DESIGN

A screw mechanism was investigated for adjusting the diameter of the device. A hex head bolt, A, is threaded into the outer leaf, 2, as shown in Figure 5. A long stud, B, is threaded into the inner leaf, 1, and fixed with respect to the inner leaf. A 3/8-16 thread was chosen for the hex head and an 8-32 thread for the stud. 16 threads per inch translates to 1.5875 mm of linear travel per revolution. 32 threads per inch translates to 0.7938

mm per revolution. When hex bolt, A, is rotated one revolution clockwise it will travel 0.7938 mm down stud, B, which is held fixed. Leaf 2 will then move up 1.5875 mm with respect to hex bolt, A. The total displacement between leaves 1 and 2 is 0.7938 mm. The prototype was designed so that the inner leaf had a thickness, t_1 , of 19.05 mm and radius, R_1 , of 55.245 mm. The outer leaf had a thickness, t_2 , of 9.525 mm and radius, R_2 , of 77.708 mm. Substituting these values into (4) yields a transmission ratio of 18.24:1. Given an 18.24:1 transmission ratio, the inner leaf will move 1/19.24 of the total travel, or 0.041 mm per revolution. On a hex head bolt, 1/12 of a rotation can be resolved. In practice this translates to rotating the hex head from a corner to the center of a flat side of the head. Thus the smallest resolvable motion of the housing is 3 μm per 30° of rotation.

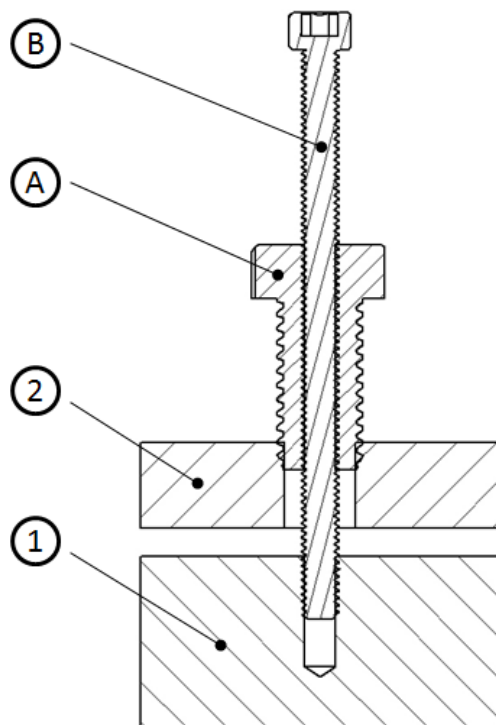


FIGURE 5. Tuning screw configuration

TESTING

A prototype section of the compliant housing was made by cutting the profile in a low-taper, tilting-head, water jet. The prototype shown in Figure 6 was made of 1 inch thick 6061 T651 aluminum. The holes on the face are for potentially stacking sections to make a longer half cylinder. The stud was threaded into leaf 1 and held fixed in that po-

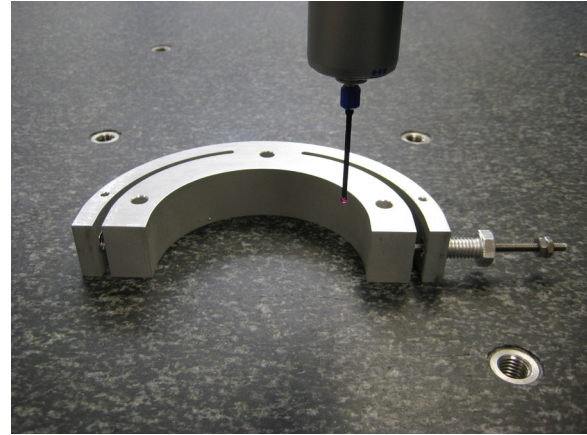


FIGURE 6. Prototype compliant housing fixtured in the CMM

sition. The hex head bolt was then threaded onto the stud and tightened until it engaged leaf 2 and was finger tight. This position was considered to be 0°. The hex head bolt was rotated in approximately 30° increments and the stud held fixed. For each position the diameter and cylindricity of the inner surface was measured with a Zeiss Eclipse CMM. Figure 7 shows the change in diameter for each 30° turn. The values are negative because the diameter was getting smaller as the bolt was turned. The horizontal axis is the starting bolt position before the turn. As can be seen from the figure there is a considerable amount of scatter in the data. This may be due to incorrectly eyeballing the 30° turn. The average change in diameter is 4 μm which matches well with the prediction of the first order approximation. Figure 8 show the cylindricity of the surface. Given that the curved beam is being loaded with a point load, it is expected that the cylindricity would get worse. However, the cylindricity changes by less than 15 μm over the course of the experiment while the diameter changed by 75 μm . These results show this system can be used to provide small adjustments to the cylinder diameter.

CONCLUSION

Two methods of making low cost cylindrical bearing surfaces were investigated. Filling behind the plastic bearing surface with epoxy yields excellent results. This method was able to replicate the diameter of a master shaft to within 2 μm while the surface held a cylindricity of 13 μm . The compliant housing provides a method to adjust the diameter of the cylindrical shape after it has been made. This device is able to change the diameter of the surface by as little as 3 μm . Future work

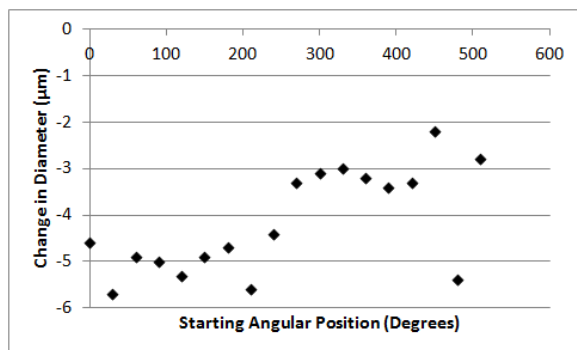


FIGURE 7. Change in diameter of the inner surface per 30° turn of the hex head bolt

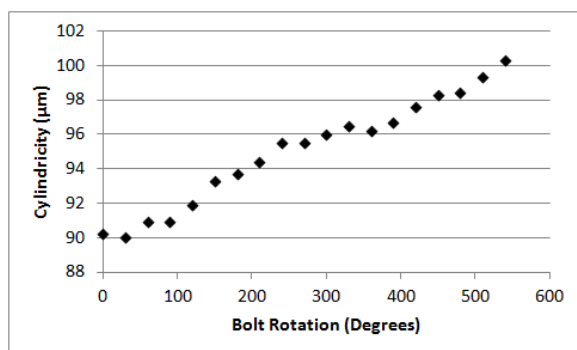


FIGURE 8. Cylindricity of the inner surface

will include combining the two techniques as well as refining the model for the compliant housing transmission ratio.

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SOLAR POWER CONCENTRATORS: BENDING METAL TUBING AT HIGH PRECISION

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INTRODUCTION

Rolling standard metal tubes is an inexpensive means of manufacturing supports for the surface elements of solar concentrators. The latest CNC tube bending machines are able to roll complex parabolic curves to approximately 0.2 – 0.3% accuracy. However, solar concentrators often require accuracy 5 – 10 times greater than that offered by a CNC bender, so we developed and tested an alternative rib forming technique.

The technique developed here involves bending individual pieces of tubing to the best possible accuracy in a CNC rolling machine to form sub-ribs, and then bonding and clamping them together to cure against a highly accurate form. This precision clamping technique results in a composite rib structure that is more accurately curved than the sub-ribs bent in the CNC tube bender.

This method was used in the design of a low cost but highly efficient parabolic cylindrical solar trough. A specific parabolic profile is required to maximize solar energy collection and metal tubes or pieces must be formed precisely to that parabolic curve in order to build the supporting ribs of the trough. This technique would be useful for other technologies as well, such as parabolic dishes.

The theoretical approach driving this technique was to maximize the ratio of the inertia of the composite rib to the total of the inertias of the sub-rib designs, in order to achieve maximum stiffness and accuracy. Simply put, the sub-rib pieces tend to spring back from the shape they are clamped to, but the high inertia of the composite structure limits this spring back, and the result is accuracy approaching that of the precision form.

We wanted to use the inertia ratios to predict the degree to which we could reduce the gap between the composite rib and the desired curvature. We use the term “reduction factor” to

describe how much the gap between the composite beam and the jig has decreased during bonding and clamping. We also developed a second model to predict the reduction factor, using the stress equations for curved beams to develop a stress ratio. We then compared the accuracy of the two models in predicting the reduction factor.

INITIAL DESIGN

A series of experiments with different composite rib designs was carried out to validate the theoretical relationship between the inertia ratio, the stress ratio and the reduction factor of the ribs.

In an approximately 1/6 scale experiment, a precision clamping jig was constructed using water-jetted plywood. For these experiments we used 6061 aluminum square pipe for the ribs, with cross section dimensions 1 inch x 1 inch x 1/8 inch thickness.

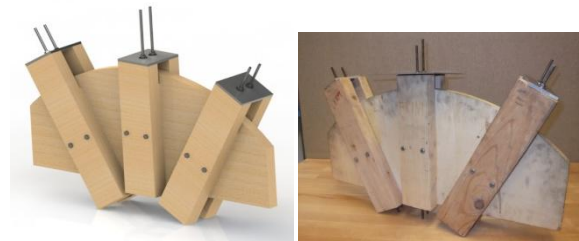


FIGURE 1. Solidworks rendering and photograph of clamping jig.

When we chose to use 6061 aluminum box extrusion as our tube stock, we calculated the maximum deflection that we could apply to the stock before exceeding the yield point of the aluminum. The maximum deflection became the maximum amount of gap that we could allow between the sub-ribs and the clamping jig before bonding and clamping the sub-ribs together. We found the maximum deflection was approximately 0.95 inches. We always bent our metal stock on the Ercolina machine so that the gap between the stock and the jig was less than the maximum allowable deflection.

We also chose an adhesive for bonding the sub-ribs together. We compared a range of Loctite products and decided to use H8600 for this application. H8600 had both high shear and peel strengths compared to the other adhesives. It also had an adequate work life of 25 minutes and was recommended for use with aluminum in severe environments, which was ideal because this solar trough is intended for use in arid desert environments.

For each experiment, we would use the Ercolina tube bender to roll 2-3 pieces of metal stock to be accurate to the desired curvature within the maximum deflection of the material as described above. We then recorded the gap between the metal tube and the clamp form. We bonded these pieces using the adhesive we selected and clamped them in our jig for 24 hours; after releasing them, we recorded the gaps again and calculated the reduction factor in the gaps before and after clamping. For most of our experiments we found that we were able to see a correlation between the reduction factor and the calculated inertia and stress ratios.

EXPERIMENTAL PROCESS

For each experiment we began by rolling our metal stock using the Ercolina tube bender, and then we applied the adhesive and bonded and clamped the sub-ribs together onto the clamping jig. When rolling the sub-ribs, we always over-rolled by a small amount – in other words, the tube was always bent to a radius smaller than the target radius (see Figure 4 below). Experiment 4b is the only exception, where the top rib was under-rolled to a radius larger than the target. In all cases the gap between the sub-ribs and the jig was less than the maximum allowable deflection.

We conducted four experiments to explore the relationship between the inertia ratio, the stress ratio and the reduction factor after bonding and clamping. Our first experiment was a simple design with two bent ribs bonded on top of one another (see Figure 2). The second design was the same except a series of “spacers” was inserted between the two sub-ribs (the spacers were 2 inch sections of a third piece of stock that we had bent). For the third experiment we used two ribs that had 0.75 inch radial cuts at 2 inch intervals, which we found relieved a lot of the stress exerted within the beam.



FIGURE 2. Experiment 1.

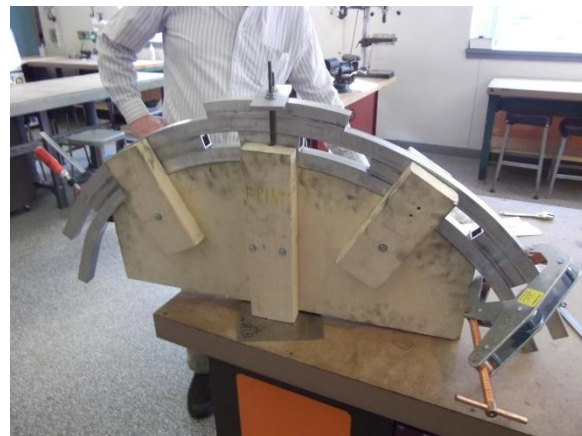


FIGURE 3. Experiment 2.



FIGURE 4. The gap between the ribs and the clamping jig before bonding is clearly visible in Experiment 3.

Our last experiment consisted of two composite ribs – the first was identical to Experiment 1. The second composite rib over-rolled the bottom rib, but under-rolled the top rib (see figures 5 and 6).



Figure 5. Experiment 4a is identical to Experiment 1.



Figure 6. Extra pieces of box extrusion were needed between the side clamps and the composite rib, because the top sub-rib was under-rolled.

Before bonding the sub-ribs we measured the gap between the ribs and the clamp. After clamping and allowing the adhesive to set for 24 hours, we repeated that measurement and calculated the reduction factor. We compared the reduction factor to the inertia ratios we calculated using both the Machinery's Handbook and using data from our Solidworks models of our ribs; we also compared the reduction factor to the stress ratios that were calculated using Matlab.

Overall we found that our inertia and stress ratios gave a good rough estimate of what the gap reduction factor for a given composite rib design would be, although the error was too large for the inertia and stress ratios to give precise predictions. The stress ratio was generally less precise than the inertia ratio, and

both ratios need to be revised before they can predict the reduction factor with acceptable accuracy.

INERTIA AND STRESS RATIOS THEORY

Through experimentation a general correlation was found between the inertia and stress ratios and the reduction factors for each composite rib design. Although the ratios do not exactly predict the reduction factor, there is a trend that as both the ratios increase the reduction factor also increases.

We used two different methods to calculate the inertia ratios for each composite rib design. The first method used Solidworks to find the section moment of inertia of the sub-ribs and the composite ribs, and then calculated the ratio between those two numbers. The inertia ratio was calculated by:

$$\text{Inertia ratio} = \frac{I_{\text{composite rib}}}{\# \text{ of sub ribs} \times I_{\text{sub rib}}} \quad (1)$$

The inertia ratios were also calculated using equations for the moments of inertia of the different sub-ribs and then using them to find the inertia ratio. These values were then compared with the Solidworks numbers and we calculated the relative error between them, which was negligible.

We used the standard equations for the stress in a curved beam to calculate the stress in the curved sub-ribs and in the composite beam. We then calculated the stress ratio as:

$$\text{Stress ratio} = \frac{\sigma_{\text{sub-rib A}} + \sigma_{\text{sub-rib B}}}{\sigma_{\text{composite beam}}} \quad (2)$$

TABLE 1. Our data from our first three experiments compared to our predicted inertia ratios.

Experiment Data with Inertia and Stress Ratios									
Exp. #	Experimental Reduction Factor	Inertia Ratio				Curved Beam Stress Ratio			
		Calculated	Error	Solidworks Value	Error	Version 1	Error	Version 2	Error
1	1.413	2.91	-106.05%	2.92	-106.65%	3.14	-122.43%	3.01	-112.91%
2	7.154	8.34	-16.58%	8.68	-21.33%	4.29	40.09%	4.15	42.05%
3	1.0156	0.17	82.91%	0.17	82.96%	3.14	-209.47%	3.02	-197.15%
4a	2.54	2.91	-14.62%	2.92	-14.96%	3.14	-23.74%	3.02	-18.82%
4b	17.67	2.91	83.52%	2.92	83.47%	3.14	82.21%	3.03	82.85%

We used two different versions of the stress ratio equation. Our first model calculated the stresses at the centroid radius of the sub-ribs and composite rib (version 1); our second model found the stresses at the inner radius (version 2). The equation used in version 1 is:

$$\text{Stress ratio} = A_{comp} r_{comp} \left(\frac{1}{A_1 r_{c1}} + \frac{1}{A_2 r_{c2}} \right) \quad (3)$$

Where:

A_{comp} = cross-sectional area of composite rib
 r_{comp} = centroid radius of the composite rib
 A_1, A_2 = cross-sectional area of individual ribs
 r_{c1}, r_{c2} = centroid radii of individual ribs

And for version 2:

$$\text{Stress ratio} = \frac{1}{A_{ind.}} \frac{1}{r_{c1} - r_{n1}} \frac{r_{i1} - r_{n1}}{r_{i1}} + \frac{1}{A_{ind.}} \frac{1}{r_{c2} - r_{n2}} \frac{r_{i2} - r_{n2}}{r_{i2}} \quad (4)$$

Where:

$A_{ind.}$ = cross-sectional area of individual rib
 r_{n1}, r_{n2} = neutral radii of individual ribs
 r_{i1}, r_{i2} = inner radii of individual ribs
 $r_{c,comp}$ = centroid radius of composite rib
 $r_{n,comp}$ = neutral radius of composite rib
 $r_{i,comp}$ = inner radius of composite rib

The results from both models were very close to one another for each experiment. The inertia and stress ratios differed by about 5% in Experiments 1, 4a and 4b, although there was about 100% error between them in Experiments 2 and 3.

CONCLUSION

Our method of bending ribs by over-rolling and then bonding and clamping them to form a composite structure is an accurate, low cost process that can be used for a variety of applications that require precise curves. An additional advantage of this method is that the inertia and stress ratios of any design can be calculated prior to construction to find an estimate of the gap reduction factor for the composite rib.

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ELEMENT LEVEL OPTIMIZATION OF A CYLINDRICAL FLEXURE FOR LINEAR MOTION GUIDANCE

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ABSTRACT

Cylindrical flexure (CF) systems have advantages over planar systems due to their geometry; however, the curvature of the CF element changes the kinematics of the flexure making current design rules insufficient for the effective description and design of these systems. The lack of relevant design rules has inhibited the design process and optimization of CF systems, preventing these systems from becoming pervasive in precision engineering applications. The first step in developing the design rules for CFs is to understand and model the behavior of the flexure elements composing the CF systems. In this paper, we compile a six degree of freedom (DOF) compliance matrix for the element and then extract from it rules on how to optimize the flexural element. The effect of two new design parameters, the sweep angle, ϕ , and the taper angle, ψ , on the flexure performance is evaluated by analyzing their effect on the element's stiffness ratios, parasitic motions, and stress distribution. The analysis is illustrated in the context of improving the performance of a CF used for linear motion guidance. Both ϕ and ψ are shown to have an effect on the performance of the flexure, resulting in two new tuning levers in the design of CF systems.

INTRODUCTION

Cylindrical Flexure Elements

We define cylindrical flexures as systems composed of flexural elements with a finite radius of curvature, R , and sweep angle, ϕ , whose length is defined by $R\phi$. The element's radial thickness is given by t_r , and its z-axis thickness by t_z . Two types of flexural elements may be constructed under the given CF definition. The element types, shown in Fig. 1, are classified by their direction of least stiffness, (A) z-compliance and (B) radial compliance. In this paper we focus on the z-compliance element.

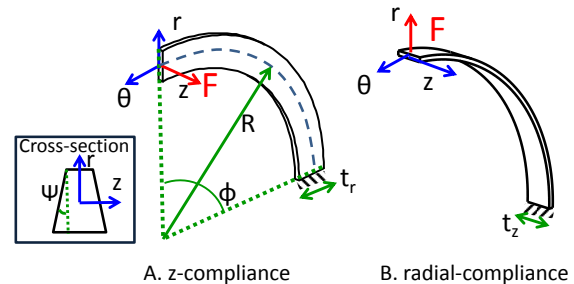


FIGURE 1. Cylindrical flexure (CF) elements are classified by their direction of least stiffness: A. z-compliance and B. radial-compliance. Length is defined by the radius of curvature, R , and the sweep angle, ϕ . The radial thickness is given by t_r and the z -axis thickness by t_z . The taper angle of the cross-section is given by ψ .

Linear Guide Attributes

We have previously presented a CF design for a linear guide for Fourier Transform Spectrometry (FTS) [2]. The original design was based on adapting the common planar compound four-bar to a cylindrical geometry. The goal of this paper is to improve on that design by optimizing the element parameters. A CF linear guide composed of z-compliance elements is characterized by its (i) range, (ii) ratio of stiffness in the z -axis direction, K_z , to the stiffness in the θ -axis, K_θ , and r -axis, K_r , directions, and (iii) ratio of displacement along the z -axis, Δz , to the two parasitic motions associated with this displacement, α_θ and α_r .

ELEMENT ANALYSIS AND OPTIMIZATION

Compliance Matrix

The foundation for the element analysis and optimization is the compliance matrix for a CF element, shown in Eq. (1). The matrix has been derived from previous works [3-4]. Each entry of the compliance matrix is factored into two components: (i) the straight beam equation and (ii) a curvature adjustment factor, ζ [1]. The matrix is presented in this manner because it allows the engineer to transfer their previous

knowledge pertaining to straight-beam flexures towards modeling the effect of the curvature. This factorization also reveals that the two parameters that cause the kinematics of the curved beam to differ from those of a straight beam are ϕ , and β . β is the ratio of the bending to torsional properties of the beam as defined in Eq. (2). The presented compliance matrix, which is the inverse of the stiffness matrix, corrects errors in previous works and has been corroborated using FEA [1].

$$\begin{bmatrix} \zeta_{11} \frac{L^3}{3EI_z} & \zeta_{12} \frac{L^3}{3EI_z} & 0 & 0 & 0 & \zeta_{16} \frac{L^2}{2EI_z} \\ \zeta_{21} \frac{L^3}{3EI_z} & \zeta_{22} \frac{L^3}{3EI_z} & 0 & 0 & 0 & \zeta_{26} \frac{L^2}{2EI_z} \\ 0 & 0 & \zeta_{33} \frac{L^3}{3EI_r} & \zeta_{34} \frac{L^2}{2EI_r} & \zeta_{35} \frac{L^2}{2EI_r} & 0 \\ 0 & 0 & \zeta_{43} \frac{L^2}{2EI_r} & \zeta_{44} \frac{L}{EI_r} & \zeta_{45} \frac{L}{EI_r} & 0 \\ 0 & 0 & \zeta_{53} \frac{L^2}{2EI_r} & \zeta_{54} \frac{L}{EI_r} & \zeta_{55} \frac{L}{EI_r} & 0 \\ \zeta_{61} \frac{L^2}{2EI_z} & \zeta_{62} \frac{L^2}{2EI_z} & 0 & 0 & 0 & \zeta_{66} \frac{L}{EI_z} \end{bmatrix} \quad (1)$$

$$\begin{aligned} \zeta_{11} &= 3\phi^3(1.5\phi - 2\sin 2\phi + .25\sin 2\phi) \\ \zeta_{12} = \zeta_{21} &= 3\phi^3(\cos \phi + (.5\sin \phi)^2 - 1) \\ \zeta_{16} = \zeta_{61} &= 2\phi^2(\phi - \sin \phi) \\ \zeta_{22} &= 3\phi^3(0.5\phi - 0.25\sin 2\phi) \\ \zeta_{26} = \zeta_{62} &= 2\phi^2(\cos \phi - 1) \\ \zeta_{33} &= 3\phi^3[\phi(1.5\phi + .5) - 2\beta \sin \phi + \sin \phi \cos \phi (.5\beta - 1)] \\ \zeta_{34} = \zeta_{43} &= 2\phi^2[\beta \sin \phi - .5(\beta + 1) + .5 \sin \phi \cos \phi (1 - \beta)] \\ \zeta_{35} = \zeta_{53} &= 2\phi^2[\beta(1 - \cos \phi + .5\sin^2 \phi) + .5\sin^2 \phi] \\ \zeta_{44} &= \phi[.5\phi(\beta + 1) + .5 \sin \phi \cos \phi (\beta - 1)] \\ \zeta_{45} = \zeta_{54} &= \phi[(\beta - 1) \cdot 5\sin^2 \phi] \\ \zeta_{55} &= \phi[.5\phi(\beta + 1) + .5 \sin \phi \cos \phi (1 - \beta)] \end{aligned}$$

$$\beta = \frac{EI_z}{GK} \quad (2)$$

In the compliance matrix, I_z and I_r correspond to area moment of inertia about the z-axis and r-axis respectively. E is the modulus of elasticity and G is shear modulus. K is the torsional stiffness constant for a given cross-section. Eq. (3) gives K for a rectangular beam, where a and b are the lengths of the two sides of the rectangle [5].

$$K = ab^3 \left[\frac{1}{3} - .21 \frac{b}{a} \left(1 - \frac{b^4}{12a^4} \right) \right], \text{ for } a \geq b \quad (3)$$

This matrix allows the engineer to design a CF using his previous knowledge of straight-beam equations supplemented with an understanding of the effect of ϕ , and β . In a CF β is important because the curvature of the element results in the application of both a bending, M_r , and twisting moment, M_θ , on the beam when F_z is applied.

β depends on the geometry of the beam, I_z and K , and the material properties of the beam, E and G . A material's E and G are related by Poisson's ratio, which is approximately 0.3 for most metals in their elastic regime. Given the majority flexures are constructed out of metal we focus our analysis on the effect of the cross-section on β . Specifically, we analyze the effect of ψ , defined in Fig. 1, on the performance of the flexure. Taper angle may be a direct result of the manufacturing method, e.g. waterjet cutting, or may be imposed during machining. The formulas for I_r , I_z , and K for a trapezoid may be found in Roark [5].

Stiffness Ratios

An important performance metric for a flexure element is the ratio of its stiffnesses. Good designs will decrease the stiffness in the desired DOF and maximize the other DOF stiffnesses. For the linear guide we aim to minimize the ratio of K_z/K_r and K_z/K_θ . A low stiffness ratio reduces undesired displacements resulting from extraneous loads.

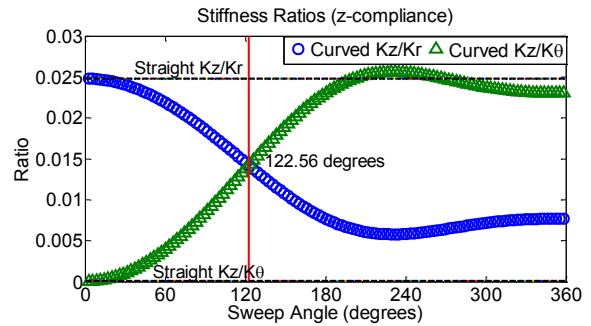


FIGURE 2. Stiffness ratios (K_z/K_r and K_z/K_θ) for a z-compliance element vs. sweep angle, ϕ . The dashed lines indicate the straight-beam ratios. The two ratios are equal at $\phi=122.6^\circ$. The length is held constant by varying R .

In the sweep angle analysis, the length of the flexure, $R\phi$, is held constant to allow us to analyze the additional role ϕ plays in determining the element kinematics. Fig. 2 shows how the stiffnesses along the direction of applied force change with ϕ ; the stiffness ratios for the curved beam are compared to those of a straight-beam of the same length and cross-section. The figure shows that ϕ gives the designer an additional lever for developing a flexure with improved F_r load rejection capability. The two stiffness ratios are equal when ϕ is

equal to 122.56° and lower than the straight beam K_z/K_r parasitic ratio.

The ψ of an element plays a role in the kinematics of the z-compliance CF, because it affects I_r and K . The optimum sweep angle of $\phi=122.6^\circ$ was used as the starting point for the taper angle analysis. Fig. 3 shows the stiffness ratios for a curved beam ($\phi=120^\circ$) vs. taper angle. The model indicates that as the taper angle increases so does the stiffness ratios, and therefore, the optimum taper angle choice is $\psi=0^\circ$. The plot shows that the straight beam kinematics is also influenced by ψ because the taper angle affects the area moment of inertia. The slope of the straight beam K_z/K_r is steeper than that of the curved beam indicating that a curved beam is less sensitive to the taper angle imposed by certain manufacturing methods.

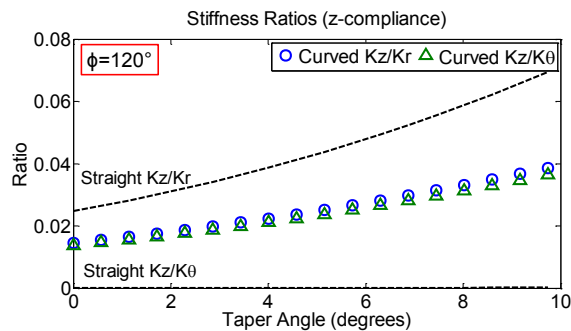


FIGURE 3. Stiffness ratios (K_z/K_r and K_z/K_θ) for a z-compliance element ($\phi=120^\circ$) vs. taper angle, ψ .

Parasitic motions

Another performance metric for a linear guide is the magnitude of parasitic displacements due to F_z compared to the desired displacement, Δz . Fig. 4 shows the ratio of parasitic, α_r and α_θ , to desired displacements, Δz , vs. ϕ . The optimal sweep angle is found to be 118° , at which point the two parasitic ratios are equal. Deviation in either direction from this point results in the increase of one of the ratios, which decreases the performance of the linear guide. The taper angle effect on the parasitic ratios of a curved beam with $\phi=120^\circ$ is less than 5% for a 10° taper angle increase.

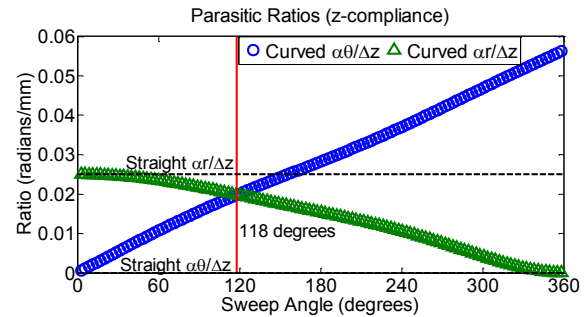


FIGURE 4. Ratio of parasitic displacements, α_r and α_θ , to desired displacement, Δz , vs. sweep angle, ϕ . The dashed lines indicate the straight-beam ratios. The two ratios are equal at $\phi=118^\circ$. The length is held constant by varying R .

Stress Distribution

The stress level imposed on a flexure during deflection determines the range of the flexural element. The stress distribution on a loaded z-compliance CF element is complex because the curvature leads to both bending and twisting moments being applied on the beam. Fig. 7 shows that the VonMises stress of a curved beam depends on ϕ . Furthermore, the figure shows the FEA calculated stress of a specific CF element normalized by the FEA value for a straight-beam of the same dimensions ($L=60\text{mm}$, $t_r=6.35\text{mm}$, $t_z=0.6\text{mm}$, $F=1\text{N}$). Eq. (5) and (6) give the values for M_r and M_θ at the base of the flexure.

$$M_r = FR \sin \phi \quad (5)$$

$$M_\theta = FR(1 - \cos \phi) \quad (6)$$

Using these two moments the bending stress, σ , and torsional shear stress, τ , on the beam may be calculated and used to find the VonMises stress, σ_v , on the beam. Fig. 7 shows the value for the calculated stress for a beam using only bending and torsional stress considerations. The plot shows that the slope of the model does not match the FEA slope when $\phi > 180^\circ$. When the sweep angle exceeds 180° the maximum σ_v no longer occurs at the base of the flexure. The model is corrected by finding the ϕ at which the max stress would occur; this angle is referred to as ϕ_{max} , and the correction is shown in Figure 7.

Fig. 7 illustrates that an analytical model based solely on the bending and torsional moments on the beam does not capture the complete effect of ϕ . Beams under torsional loading with a fixed-end experience additional longitudinal stresses

due to the elongation of the outer fibers during twisting and warping [5]. For a rectangular beam of closed cross-section, these warping stresses are usually small compared to the torsional shear stresses. However, in a z-compliance CF element these longitudinal warping stresses are significant because they act in conjunction with the bending stresses on the beam. A parametric study was performed to determine an analytical model for these warping stresses. The warping stress models for thin-walled open cross sections were used as the basis of the model. A preliminary analytical expression for the warping stress of the CF element is given in Eq. (7) [5]. Fig. 7 shows the stress calculated by adding the warping stress to the bending stress in the VonMises equation.

$$\sigma_{warp} = .61 \left(\frac{E}{G} \right)^{0.5} \tau \left(1 + \frac{t_r}{100t_z} \right) \tanh \phi_{max} \quad (7)$$

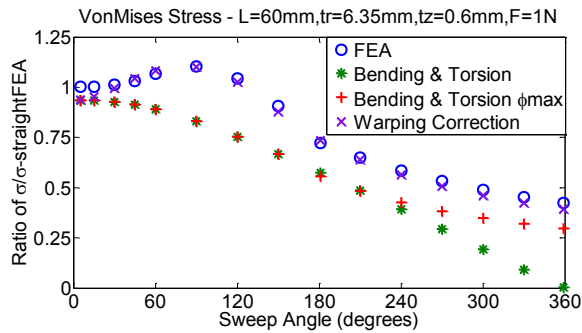


FIGURE 5. VonMises stress vs. sweep angle, ϕ , normalized by the FEA value for a straight beam. The FEA values, 'o', are compared to the values calculated by different stress models, and the presented warping model is given by 'x'.

The maximum range of the flexure may now be calculated using the presented warping correction stress model, the compliance matrix, and the yield stress for a material. Future work will expand the stress model to account for the taper angle effect and will also determine the application limits for this model.

PROTOTYPE REDESIGN

The sweep and taper angles of a curved-beam element have been shown to serve as two additional parameters the engineer can tune to optimize the performance of a CF system. The performance of the single DOF linear guide was optimized by minimizing the stiffness ratios and parasitic motions. Using the presented analysis, the FTS linear guide was redesigned to be formed by elements with $\phi=120^\circ$ and $\psi=0^\circ$. The

new design uses only two compound four-bars to maintain the same cylinder length. The stress model was used to determine the radius of the CF given the desired 24 mm range. Fig. 8 illustrates the old design and new design. Table 1 compares the performance of the two designs as calculated using FEA.

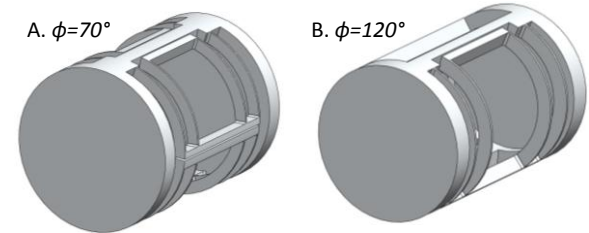


FIGURE 8. FTS Linear guide designs: A. original $\phi=70^\circ$; B. element optimization redesign $\phi=120^\circ$

TABLE 1. FEA performance comparison of original prototype ($\Phi=70^\circ$) and new design based on element optimization ($\Phi=120^\circ$).

	Original $\phi=70^\circ$	New $\phi=120^\circ$
Radius	41.3mm	31.0mm
$K_z/K_r, K_z/K_\theta$	$1.04e^{-2}, 2.35e^{-2}$	$1.17e^{-2}, 6.43e^{-2}$
$\alpha_r/\Delta z, \alpha_r/\Delta z$ (rad/mm)	$9.08e^{-5}, 1.23e^{-2}$	$3.46e^{-5}, 3.46e^{-4}$

Table 1 demonstrates that both parasitic ratios are improved in the new design, while the stiffness ratios are larger but remain of the same order of magnitude. The unexpected stiffness ratios results indicate that the effect of the layout and number of elements needs to be analyzed. The next step in the design of CF systems is to study how the curvature of the elements changes the system creation rules. Future work will extract design rules towards accurately assembling and optimizing CF systems.

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INVESTIGATION OF ULTRA-PRECISION TURNING USING AEROSTATIC AND HYDROSTATIC SPINDLES

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INTRODUCTION

The continuous progress in technology and miniaturization of optical components requires ultra-precise machining of ferrous and non-ferrous molds and dies.

Aerostatic spindles, capable of reaching high rotational speed, are conventionally used to turn non-ferrous materials like NiP using mono-crystalline diamond (MCD) tool.

However, for turning ferrous materials like hardened steel with a cubic boron nitride (CBN) tool, the aerostatic spindle lacks the necessary damping to overcome the high loads formed during hard turning.

One alternative to tackle this problem is the use of an oil hydrostatic spindle which effectively combines ultra-high rotational accuracy, high static stiffness and load capacity, extremely high vibration resistance and moderate thermal expansion [1].

The other alternative is to cover the work-tool with Plasticine and therefore increasing the damping of the system during turning.

In this paper hard turning and diamond turning using an aerostatic and a hydrostatic spindle are investigated. Additionally, the effective use of Plasticine in increasing the system damping is demonstrated.

EXPERIMENTAL PROCEDURE

Test benches

The experiments are conducted at three test benches: MMC 1300, Precitec Nanoform 250 ultra and MTC 410 Lathe. Both the MMC 1300 and the Precitec are equipped with an aerostatic spindle, whereas the MTC 410 Lathe is equipped with an oil hydrostatic spindle.

The experiments vary from plane diamond turning of a NiP galvanically coated steel cylinder having an outer radius $r_o = 40$ mm and an inner $r_i = 5$ mm, to plane hard turning of a hardened steel cylinder with $r_o = 25$ mm and $r_i = 5.5$ mm. Each cylinder's surface is divided

equally into 4 zones (fig.1). This will help investigating the influence of the cutting speed variation on the surface finish.

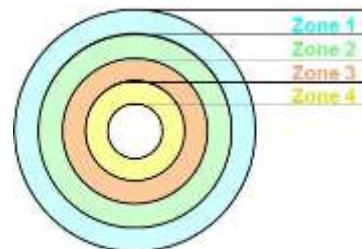


FIGURE 1. Different zone distribution on the work piece surface.

Precitec Nanoform 250 ultra

On the Precitec test bench, an aerostatic spindle with a maximum rotational speed $n = 7000$ rpm is used for plane hard turning of a 50 mm diameter cylinder made of steel 1.2083 hardened to 53 HRC. The cutting tool is a CBN from the company Chardon Tool.

Two process parameters are then varied. In the first experiment the cutting speed is kept constant, whereas in the second experiment the spindle speed is kept constant (table 1).

UP MMC 1300

The MMC 1300 test bench comprises an ISO 55 aerostatic spindle from Professional Instruments Company that can reach a maximum rotational speed of $n = 10000$ rpm. It is used to turn an 80 mm diameter NiP coated steel cylinder with an MCD cutting tool under constant rotational speed of $n = 1500$ rpm. The turning process parameters are shown in table 2.

MTC 410 Lathe

The MTC 410 Lathe is equipped with a bi-conic 4RH oil hydrostatic spindle from Professional Instruments Company (fig. 2) and can reach a maximum rotational speed of $n = 2000$ rpm due to amplifier limitations. The spindle itself runs up

to 3600 rpm. The spindle is equipped with a System 3R chuck for work-piece fixation. The MTC Lathe is used for plane diamond turning and hard turning of the work-pieces mentioned respectively for the UP MMC 1300 and Precitec test benches. The process parameters are shown in table 1 and table 2.



FIGURE 2. Bi-Conic 4RH oil hydrostatic spindle mounted on an MTC 410 turning machine.

RESULTS

The results are compared between aerostatic and hydrostatic spindles based on approximately the same process parameters and divided into two sections: hard turning of hardened steel and diamond turning of NiP.

Hard Turning

Hardened steel is turned on the Precitec under a constant cutting speed $V_c = 80$ m/min, a feed rate $f = 4.8$ mm/min and a depth of cut $a = 10\mu\text{m}$ using a CBN work tool with a radius $r = 0.75$ mm. A non-uniform distribution of the average roughness R_a is observed along the machined surface when measured with Taylor Hobson tactile device (fig. 3, top). The highest value of R_a is found at the periphery of the work-piece (zone 1) where the spindle vibration reaches the strongest value as the cutting occurs at the farthest distance from the spindle z-axis. Other factors like the relative vibration between the tool and the work-piece or the tool tip vibration can cause a decrease in the surface quality of the machined piece.

One simple but effective solution to tackle this problem is to increase the damping of the work tool by covering it with Plasticine (fig.4).

A clear decrease in the average roughness value and a uniform roughness distribution along the surface is then observed (fig. 3, bottom).

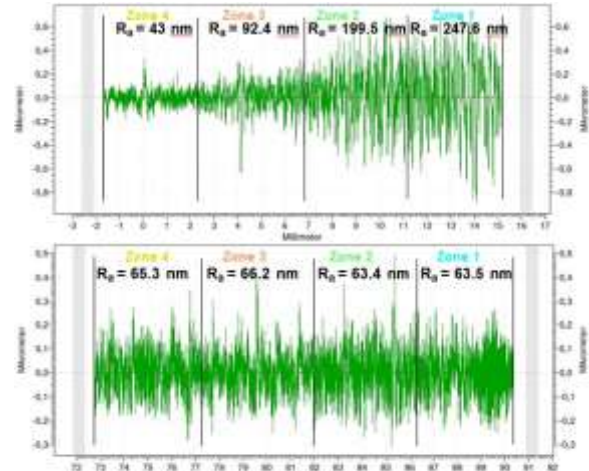


FIGURE 3. Average roughness of a hardened steel work-piece turned on Precitec at $V_c = 80$ m/min without Plasticine (top) and with the use of Plasticine (bottom).



FIGURE 4. Plasticine covering the work-tool.

Use of Plasticine

During hard turning, the cutting forces are expected to be relatively high and unsteady due to chip formation. This causes sharp peaks in the cutting force creating high frequency alternative force applied between the work-tool and the work-piece [1]. As a result, bad finish surface is observed when measuring the surface average roughness R_a using a white light interferometer WYKO (fig.5, left) in comparison to a surface turned while covering the tool with Plasticine (fig.5, right).

Since it is a non-drying oil-based material, Plasticine helps decreasing the vibrations created during turning and to smoothen sharp peaks in the cutting force.

The use of Plasticine is particularly effective when turning with an aerostatic spindle because the latter usually lacks of high damping in comparison to an oil hydrostatic spindle.

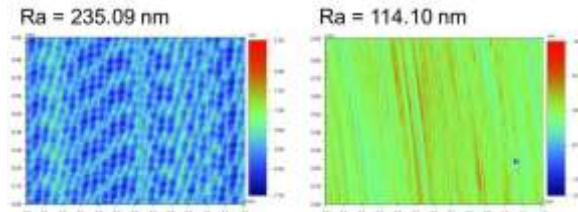


FIGURE 5. Average roughness R_a of a hardened steel surface turned using an aerostatic spindle without (left) and with (right) Plasticine covering the work-tool.

Along this work, Plasticine will be used to cover the work-tool for all experiments conducted with aerostatic and hydrostatic spindle for diamond turning and hard turning.

Hardened steel is now turned on Precitec and MTC while keeping the respective spindle rotational speed constant (table 1).

The average roughness value is not uniform in both spindles and mainly increases as the cutting speed decreases close to the center of the machined piece (zone 4, fig.7 and 8).

TABLE 1. Process parameters used to turn hardened steel on UP Precitec and MTC Lathe.

Process parameters	UP Precitec	MTC 410 Lathe
Spindle speed n	2425 rpm	1600 rpm
Cutting speed	variable	variable
Feed rate f	4.8 mm/min	4.8 mm/min
Depth of cut a	10 μ m	20 μ m
Work tool type	CBN	CBN
Work tool radius	0.75 mm	0.75 mm
Average roughness	40.33 nm $\leq R_a \leq$ 104.55 nm	41.64 nm $\leq R_a \leq$ 76.49 nm

However when directly comparing the surface roughness obtained on MTC with the one on Precitec, the former shows a decrease in roughness although the depth of cut was doubled compared to the value used on Precitec (table 1).



FIGURE 6. Hardened steel cylinders turned on Precitec (left) and MTC (right).

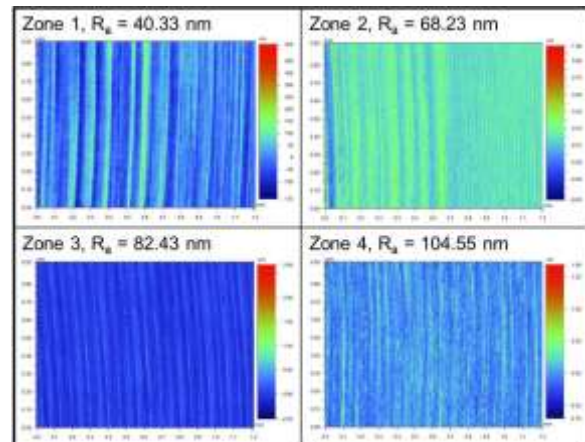


FIGURE 7. R_a measured with WYKO of a hardened steel surface turned on Precitec.

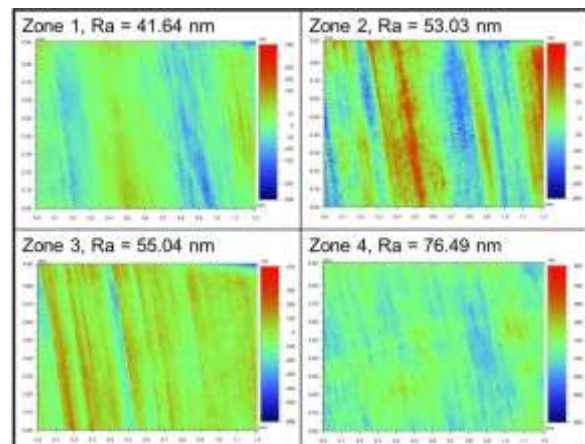


FIGURE 8. R_a measured with WYKO of a hardened steel surface turned on MTC.

Diamond Turning

Nickel Phosphorous is then diamond turned on the MMC 1300 and the MTC 410 Lathe under a constant spindle speed. An additional advantage of the damping characteristics of the oil hydrostatic spindle is the possibility to increase productivity of the cutting process [3]. Sub-nanometer R_a values are known in direct MCD machining with a submicron feed rate and single digit micron infeed causing tremendous machining time [4]. For the following investigation, the cutting parameters for rough diamond machining have been adjusted in order to analyze the surface finish result. The measurement of the average roughness using WYKO indicates a decrease of more than 50% in value for the work-piece turned with the hydrostatic spindle (MTC 410, fig.9) in comparison to the one turned using the aerostatic spindle (MMC 1300).

The process parameters and the roughness values are shown in table 2.

TABLE 2. Process parameters used to turn NiP on MMC 1300 and MTC Lathe and the respective average roughness.

Process parameters & results	UP MMC 1300	MTC 410 Lathe
Spindle speed n	1500 rpm	1500 rpm
Cutting speed	variable	variable
Feed rate f	15 mm/min	15 mm/min
Depth of cut a	10 μm	10 μm
Work tool type	MCD	MCD
Work tool radius	1.011 mm	1.012 mm
Average roughness R_a	$3.5 \text{ nm} \leq R_a \leq 4.5 \text{ nm}$	$1.3 \text{ nm} \leq R_a \leq 2.1 \text{ nm}$

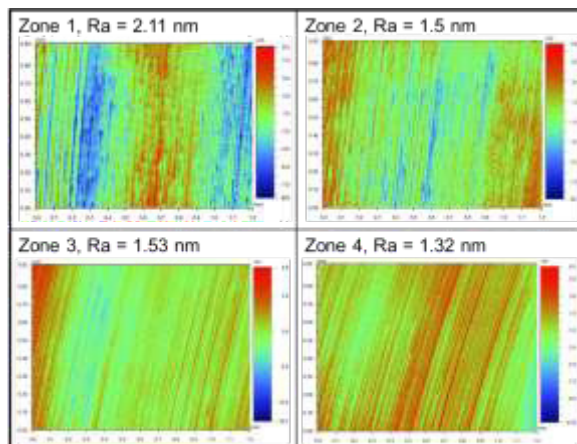


FIGURE 9. R_a measured with WYKO of a NiP coated surface turned on MTC 410 Lathe.

Hydrostatic Spindle Thermal Growth

A disadvantage of the oil hydrostatic spindle is the higher friction compared to the aerostatic spindle and along with it the thermal growth. The thermal expansion of the hydrostatic spindle is measured using a capacitive sensor [2] mounted externally along the spindle z- axis.

Starting from cold state, the spindle is then set for a rotational speed $n = 1500 \text{ rpm}$ turning CW for a period of 3 hours and 20 min. (12000 s).

The near-logarithmic thermal expansion of the hydrostatic spindle reaches a constant value of $8 \mu\text{m}$ after 2 hours and 20 min. (8400 s, fig.10). For good form accuracy a pre-run of the spindle is essential. Further investigation will be dedicated to improving the oil chilling and heating respectively.

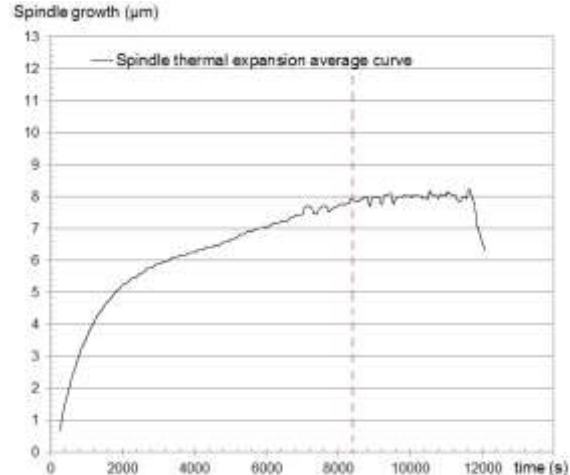


FIGURE 10. Plot of the thermal expansion of the hydrostatic spindle in the z- axis versus time.

CONCLUSION AND OUTLOOK

Optic surface finish has been achieved when diamond turning of NiP on a hydrostatic spindle using low spindle speed ($n = 1500 \text{ rpm}$) and high feed rate ($f = 15 \text{ mm/min}$). This is of particular importance in the production where time plays a decisive role. Average roughness values of less than 2.1 nm have been as a result measured on the work-piece surface.

The hydrostatic spindle did effectively combine good surface finish ($41.64 \text{ nm} \leq R_a \leq 76.49 \text{ nm}$) with time efficiency when machining hardened steel with moderate depth of cut ($a = 20 \mu\text{m}$) and feed rate ($f = 4.8 \text{ mm/min}$), and under a constant spindle speed.

As future works, work-pieces will be turned on the MTC under constant cutting speed where better surface finish is expected.

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FINISHING OF ELECTROLESS NICKEL PLATED DIES FOR HARD X-RAY MIRROR REPLICATION

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INTRODUCTION

X-ray telescopes occupy a unique niche in astronomy, allowing astronomers to understand the distribution of hot gas between galaxies, the physics of supernova remnant explosions, black holes radiation, and other important issues [1]. Recent X-ray space telescopes such as Chandra [2], XMM-Newton [3] and Suzaku [4] were launched between 1999 and 2005. Each of these three telescopes were constructed from multiple nested grazing incidence reflecting mirrors consisting of a paraboloid and hyperboloid of revolution, in a configuration known as Wolter type-1 [5] (see Figure 1, right). These telescopes were targeted at low-energy “soft” X-Ray emissions, which require a surface roughness of 0.5nm rms or less.

Plans for the next generation of X-ray telescopes, amongst which the ASTRO-H project, aim at focusing high-energy “hard” X-ray radiations, up-to 80keV. For such radiations to be reflected correctly, the specification of the nested mirrors becomes more stringent. The surface texture requirement is 0.3nm rms or less, and global form error less than 100nm P-V. Since ultra-precision diamond turning is well known to deliver very accurate form on turnable

materials, such as aluminum and nickel, a possible manufacturing method of thin Wolter type-1 mirrors for hard X-ray space telescopes involves generating electroless nickel plated mandrels by diamond turning, before coating them with a reflective film and substrate [6] (see Figure 1, left).



FIGURE 2. Generation of X-ray replication mandrel on large diamond turning machine.

However, because of the large size of the outer mandrels (up-to 600mm diameter, see Figure 2) the distance travelled by the diamond tip during a single pass may reach dozens of kilometers.

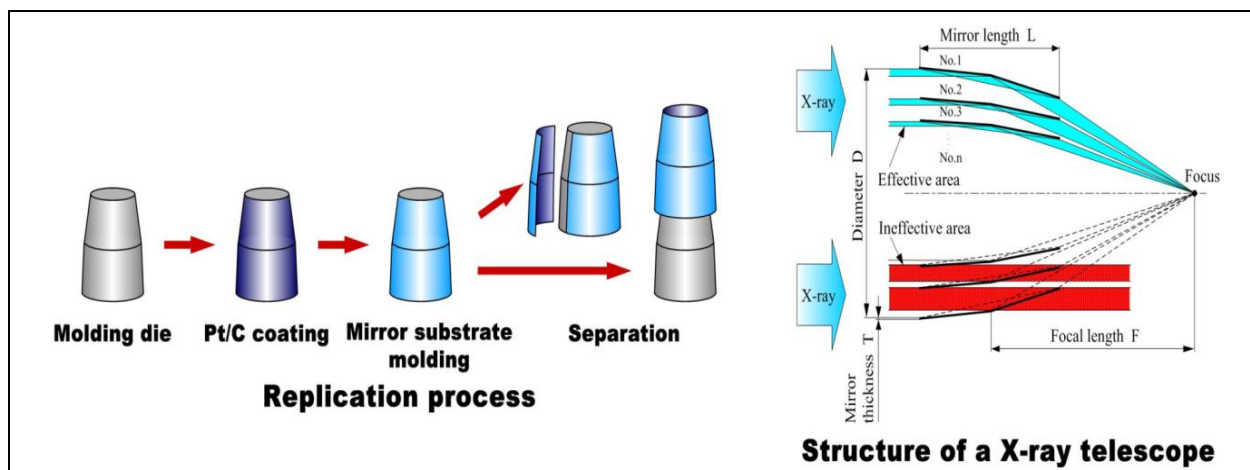


FIGURE 1. Proposed method for fabricating Hard X-ray telescopes by replication from moulding dies.

Tool wear occurring during such a pass is thus usually extensive, in the order of 1 micron or more [7]. As there is currently no ability to sharpen the tool during a pass, the tool wear causes a broadening of the cutting edge and thus of the turning marks. The surface texture of the turned mandrels thus falls far short from the requirements of both X-ray and EUV applications (see Figure 3). The high-frequency machining marks need to be removed in order to prevent adverse effects on the optical performance (such as diffraction and scattering). There is thus a compelling need for an automated finishing method capable of smoothing diamond turned surfaces down to 0.3nm rms or less.

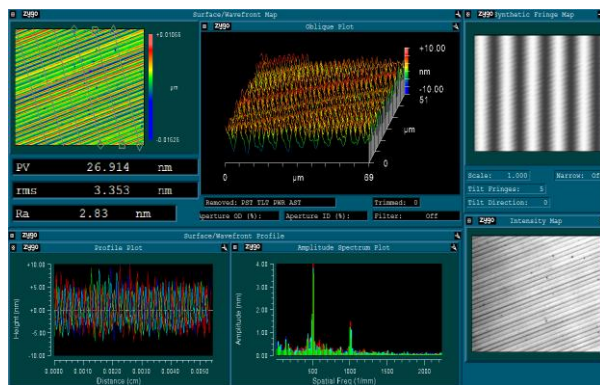


FIGURE 3. Typical surface texture measured after diamond turning operation (3.3nm rms).

To meet this challenge, a two-step finishing method is presented that combines fluid jet and precessed bonnet polishing on a common 7-axis CNC platform (see Figure 4). This hybrid method is capable of smoothing diamond turned electroless nickel plated dies from 3.0nm rms down to 0.28nm rms roughness, while deterministically improving form error down to 30nm P-V [8].



FIGURE 4. Fluid jet and precessed bonnet polishing on common 7-axis CNC platform.

FLUID JET POLISHING

The fluid jet polishing process consists of pressurized water and abrasive particles mixed

and delivered through a nozzle [9]. Typical operating pressures range from 4 to 20bar, while typical abrasives range from 0.1 to 100um. This process was recently improved with a redesigned slurry delivery unit [10] (see Figure 5). This new system comprises 2 slurry pumps (low and high pressure), a pressure feedback loop, and an adjustable bypass valve for regulating flow rate conditions inside the system.

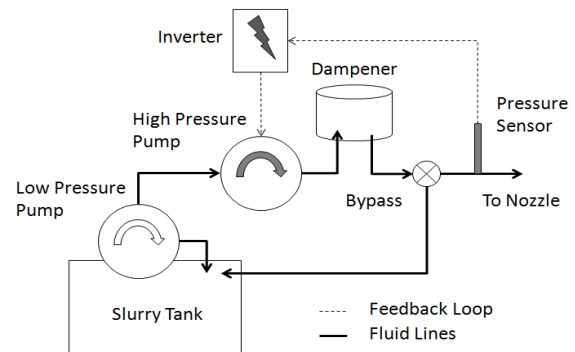


FIGURE 5. Diagram of improved slurry delivery system to fluid jet nozzle.

A turbulent fluid dynamics simulation was also generated, in order to characterize and optimize the stability of the fluid jet stream via adjustments of the feedback loop parameters and bypass valve. This optimization was aimed at neutralizing waviness effects induced by removal rate instability during polishing. To validate the optimization procedure, small samples of optically polished BK7 glass were re-polished by a raster tool path with 1.5um cerium oxide particles, both before and after performing the optimization with the numerical model.

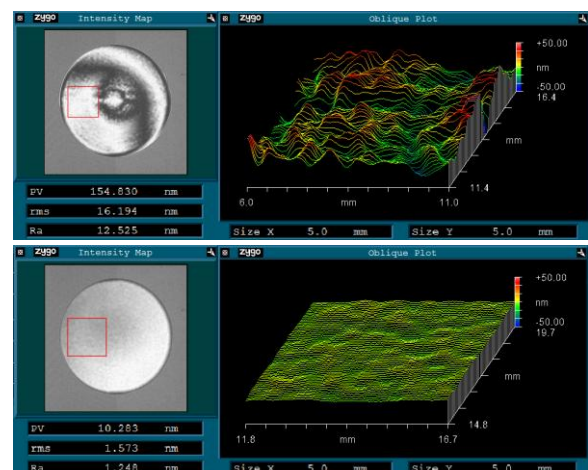


FIGURE 6. Surface texture after polishing BK7 with 1.5um cerium oxide (top) before and (bottom) after optimization by numerical model.

The resulting surface texture measurements showed significant reduction in the level of waviness introduced by the fluid jet polishing on the surface, from 16nm rms down to 1.5nm rms (see Figure 6).

In the case of the diamond turned X-ray mandrels, 1.2 μ m aluminum oxide particles were mixed with pure water at a specific gravity of 1.06 and pressurized at 16 Bar through a 2 mm diameter nozzle. It was possible to polish the X-ray mandrels with a spiral tool path circling the surface from top to bottom. Measurement of the fluid jet polished surface revealed a complete removal of the diamond turning marks, together with a lowering of the surface texture from 3.35nm rms down to 2.1nm rms (see Figure 7). Most importantly, the surface signature after fluid jet polishing was random instead of periodic, as can be seen in the amplitude spectrum plot.

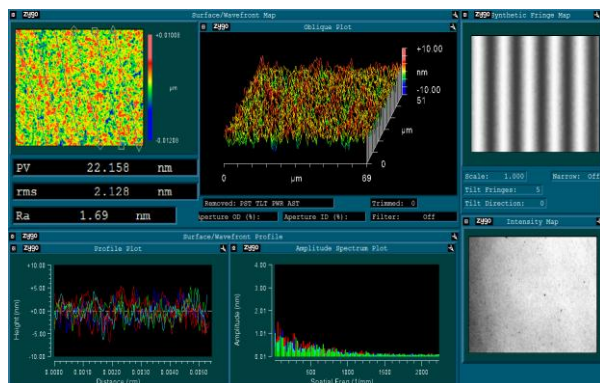


FIGURE 7. Typical surface texture measured after fluid jet polishing operation (2.1nm rms).

BONNET POLISHING

Following on from the fluid jet polishing step, the surface can be brought to final super-smooth finish by use of the precessed bonnet polishing method [8].

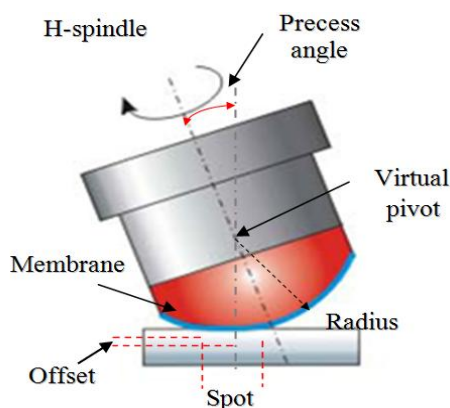


FIGURE 8. Principle of precessed bonnet tool.

The operation of this process may be summarized as follows: The position and relative orientation (precess angle) of a spinning, spherical tool are actively controlled to generate a sub-aperture polishing spot area on the work-piece surface (see Figure 8). But while a classical polishing tool is pressurized against the surface of the work-piece, with no attempt to control actively the Z position of the tool in a local or global coordinate frame, in the technique we describe the Z position and precess (but not directly the contact-force) are actively controlled with a CNC machine tool.

Rather than relying on the usual “discreet precess” polishing regime described in previous literature, whereby the tool is precessed in a given direction for each polishing pass (see Figure 9, left), a novel tool pathing method called “continuous precessing” is presented in this paper. In this method, the direction of the surface tangent used to compute the plane of precession of the spherical tool is allowed to spin around the centre of the polishing spot (see Figure 9, right).

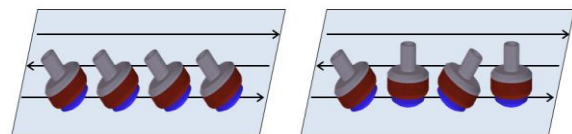


FIGURE 9. Diagram of (left) discreet precess mode, and (right) continuous precess mode

This motion may be compared to the wobbling of a spinning top, or the progressive change in the rotation axis of the earth: in those two examples the rotation axis describes a cone. In a similar way, the precess vector describes a cone around the polishing spot thereby allowing the polishing direction to change continuously.

To assess the benefits of this method, small samples of laser grade fused silica glass were re-polished with a raster tool path and 1.5 μ m cerium oxide particles, in both discreet and continuous precess mode. The surface feed rate was kept constant at 200mm/min, with raster track spacing of 0.1mm, tool offset of 0.2mm, and precess angle of 20deg. The resulting surface texture measurements showed significant improvement in the qualitative level of directionality of the polishing marks (see Figure 10). It was also noted that the overall roughness was improved by the new method, from 1.5nm rms down to 0.6nm rms.

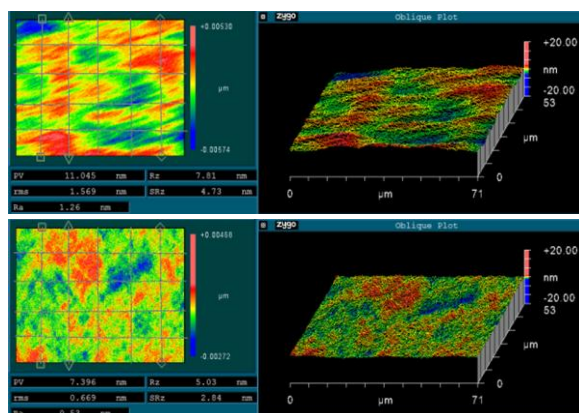


FIGURE 10. Surface texture after polishing fused silica with 1.5 μm cerium oxide using (top) discreet and (bottom) continuous precess mode.

In the case of the fluid jet polished X-ray mandrels, very fine slurry of 7nm fumed silica particles mixed with pure water at a concentration of 30g/L was fed above the bonnet with a disposable pipe connected to a peristaltic pump. To prevent drying and crystallization of the slurry on the work-piece surface, a series of pure water atomizers were arrayed around the bonnet to keep a high humidity level inside the polishing enclosure. The mandrels were then polished with a spiral tool path circling the surface from top to bottom, while the bonnet tool performed the continuous precessing motion. This brought the final surface texture down to 0.28nm rms (see Figure 11).

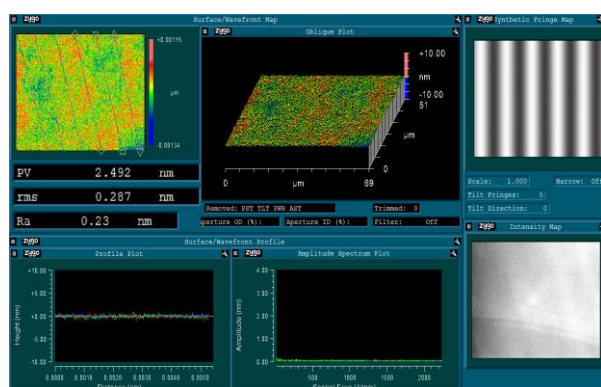


FIGURE 11. Surface texture of X-ray mandrel after continuous precess finishing (0.28nm rms).

CONCLUSION

Using a two-step process consisting of fluid jet polishing followed by precessed bonnet polishing, it is possible to bring the surface texture of diamond turned electroless nickel mandrels from circa 3.0nm rms down to less than 0.3nm rms, as required for hard X-ray applications.

This result was achieved by improving the stability of the fluid jet process to make it free from waviness, and using the new continuous precessing polishing mode to deliver a final surface texture free from any directionality, a critical factor with X-ray optics.

It should be noted that the results from this study can also be extended to soft X-ray and EUV optics, thus opening the door to more efficient finishing of replication mandrels for applications in these other radiation wavelengths.

ACKNOWLEDGMENTS

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A STUDY OF THEORETICAL MODELLING OF SURFACE GENERATION IN COMPUTER CONTROLLED ULTRA-PRECISION POLISHING OF 3D-STRUCTURED SURFACES

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INTRODUCTION

Over the past decades, surfaces possessing specially-designed 3D-structures are widely used in the development of products such as micro-lens array, v-grooves, etc. Moreover, some research work [1] shows that a surface with a longitudinal rib structure has a lower shear stress than a smooth surface. This technology can be used in commercial aircraft [2]. In nature, compound eyes and shark skin are typical examples of 3D-structured surfaces. Although there has been extensive research on different types and scale of 3D-structured surfaces, the research work on fabrication and machining of these 3D-structured surfaces is far from complete.

Computer Controlled Ultra-precision Polishing (CCUP) is capable of fabricating ultra-precision freeform surfaces with sub-micrometer form accuracy and nanometer surface roughness. The CCUP possesses technological merits over the conventional multi-axis CNC mechanical machining such as micro-milling or grinding. These include: (a) CCUP can process hard-to-machine materials and brittle materials; (b) High surface accuracy with submicrometer form error and surface finish in nanometer range; (c) Less susceptible for tool wear; (d) Different structure surfaces can be generated by modifying tool influence function instead of changing cutting tool with different geometries.

CCUP is a complex process and our understanding of the surface generation and the factors affecting the surface generation of CCUP is still far from complete. Most of the previous research focuses on experimental and empirical studies. There is a need for the development of more economic and controllable methods to construct structured surfaces. As a result, this paper presents a theoretical analysis of surface generation in CCUP of 3D-structured surfaces.

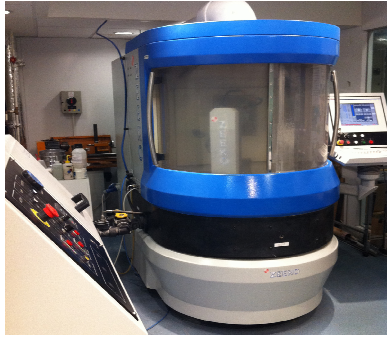
COMPUTER CONTROLLED ULTRA-PRECISION POLISHING (CCUP)

Computer controlled ultra-precision polishing (CCUP) is an emerging technology is one type of the abrasive machining processes. The cutting is based on either a bonnet tool or a pressured fluid jet. The metal removal is accomplished by a multi-point cutting process with the interaction of the polishing tool, slurry and workpiece. The fixed workpiece is usually machined with a rotating polishing head with multi-axis motions controlled by computer numerical controller.

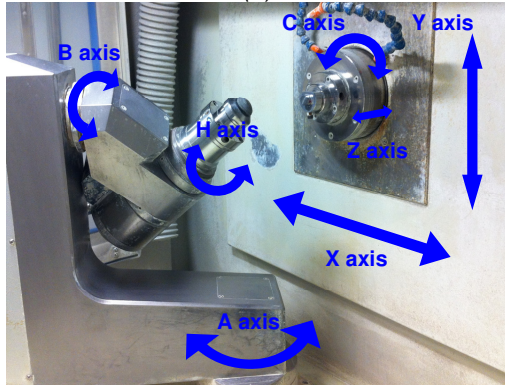
In the present study, the CCUP is conducted based on a 7-axis freeform polishing machine named Zeeko IRP200 from Zeeko™ Ltd., UK as shown in Fig. 1. The machine has a 7-axis mechanism of motion of which four axes control the workpiece motion and the other three axes control the polishing head.

As shown in Fig. 2(a), the mechanical polishing makes use of a plastic bonnet covered with a polishing cloth for performing the polishing process. Fig. 2(b) shows the polishing geometry for the mechanical ultra-precision polishing. The factors affecting the surface generation in mechanical polishing are basically composed of process factors and materials factors. The former include surface feed, precess position, precess angle, head speed, head pressure, tool radius, tool offset, tool overhang, workpiece radius, etc. The latter includes cloth material, slurry medium, workpiece material, slurry temperature, slurry specify gravity, etc.

As as shown in Fig. 3, the surface quality is also affected by the polishing strategies which include spiral, raster, spokes, and random modes, etc.

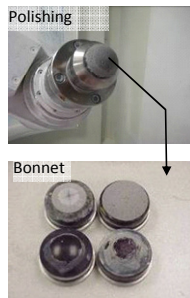


(a)

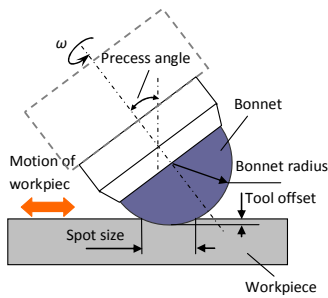


(b)

Figure.1 Zeeko IRP200 Ultra-precision freeform polishing machine

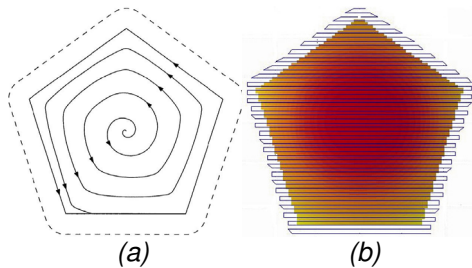


(a)



(b)

(a) Polishing tools; (b) Polishing geometry
Figure 2. Mechanical polishing of CCUP



(a)

(b)

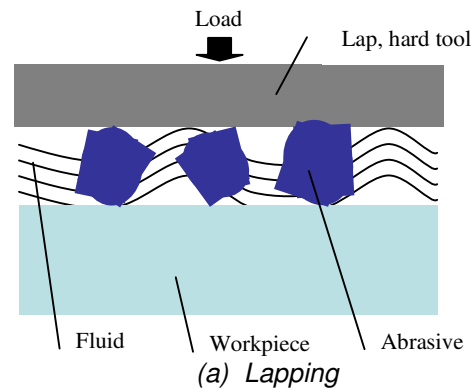
Figure 3. Polishing strategies: (a) Spiral and (b) Raster

MATERIAL REMOVAL MECHANISMS IN POLISHING PROCESS

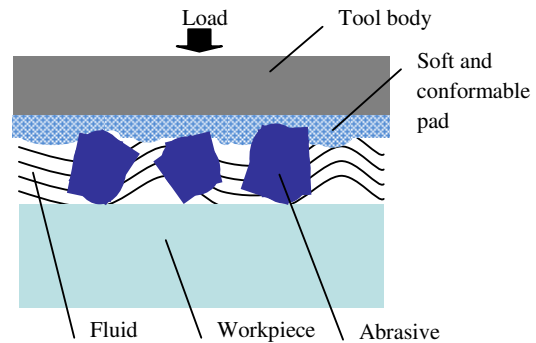
Polishing is one type of the abrasive machining processes which makes use of abrasive particles to modify the surface finish and form accuracy of the workpiece. Generally, the abrasives (polishing particles) are introduced between the polishing tool and the workpiece in both lapping and polishing. The difference is that lapping makes use of hard tool while polishing uses soft tool as shown in Fig. 4. Fig. 5 shows the material removal mechanisms for polishing which include micro-cutting, micro-chipping, micro-cracking, scratching and ploughing, etc. Assuming that the geometrical shape of the abrasive particles can be simplified as spherical or conical form, then the material removal rate in terms of mass can be expressed as [3]:

$$MRR_{\text{mass}} = \rho_w \cdot N \cdot Vol + C_0 \quad (1)$$

Where, ρ_w is the density of the workpiece, N is the number of the active abrasive particles, Vol is the volume removed by a single abrasive particle per unit time, and C_0 is the material removal due to chemical etching.



(a) Lapping



(b) Polishing

Figure 4. Lapping and polishing principles.

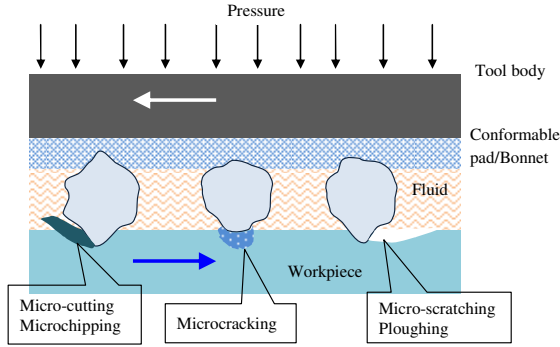


Figure 5. Polishing mechanism.

The key problems for the model presented by Eq.(1) are the ways for the determination of number of the active abrasive particles (N) and the volume removed by a single abrasive particle (Vol), by neglecting the chemical etching effect when the solid-solid contact is applied for the polishing process.

THEORETICAL ANALYSIS OF SURFACE GENERATION IN CCUP OF 3D-STRUCTURE SURFACES

In the present study, a surface topography simulation model is developed based on the determination of Tool Influence Function (TIF), which indicates the tool material removal characteristics. The theoretical base of the surface topography simulation model is focused on the modeling of the TIF.

As a beginning, Preston's Law [4] is firstly employed to establish the TIF, which is relative to the polishing tool pressure on the surface of the workpiece and the relative speed between the tool and the workpiece. According to Preston's Law, the Integrated Material Removal Rate (IMRR) is

$$M_p = k \cdot P_p \cdot V_p \cdot t_d \quad (2)$$

Where, M_p is the material removal at point P , k is Preston coefficient, P_p is polishing pressure, V_p is the magnitude of relative speed at point P , t_d is dwell time.

As a result, MRR at point $P(\alpha, \theta)$ is

$$M_p(\alpha, \theta) = k \cdot P_m \exp\left(-\frac{1}{2}(D \tan(\alpha)/b)^2\right) \cdot \left(\left(\|\mathbf{v}_r\|v_{rx} + \|\mathbf{v}_f\|v_{fx}\right)^2 + \left(\|\mathbf{v}_r\|v_{ry} + \|\mathbf{v}_f\|v_{fy}\right)^2\right)^{\frac{1}{2}} \cdot t_d \quad (3)$$

Eq. (3) is the polishing tool material removal characteristics, or influence function.

In the surface topography simulation model, a Modified Gaussian Function (MGF) is employed to construct the TIF in CCUP. The tool influence function (TIF) for polishing is determined according to the theoretical and experimental results. Hence, the surface generation of the structured surfaces is predicted and simulated.

PRELIMINARY RESULTS AND DISCUSSION

To proof the concept for the surface topography simulation model, preliminary simulations and polishing experiments have been conducted. Table 1 shows the polishing parameters. The workpiece is made of nickel copper and is a cylinder (diameter: 10 mm) with a flat surface. Figure 6 shows the Tool Influence Function (TIF) in (a) two dimensions, (b) in YZ plane and (c) in XZ plane, generated based on the provided parameters.

Figure 7 shows the experimental results. It is interesting to note that the predicted surface pattern agrees well with the measured results. The results of the pilot study not only provide an important means for making the CCUP to be more predictable but also help to minimize the costly and time consuming trial and error experiments to establish the optimum polishing conditions.

TABLE 1. Polishing parameters of polishing test.

Mechanical Polishing	1 st Polishing (Vertical)	2 nd Polishing (Horizontal)
Polishing mode	Raster	Raster
Precess angle (deg)	15	15
Head speed (rpm)	1200	1200
Tool offset (mm)	0.3	0.3
Tool pressure (bar)	1	1
XY spacing (mm)	1	3
Surface feed (mm/min)	50	50
Tool size	20mm radius bonnet	20mm radius bonnet
Polishing cloth	Cerium oxide D'16	Cerium oxide D'16
Polishing slurry	Al2O3 (1.5μm)	Al2O3 (1.5μm)

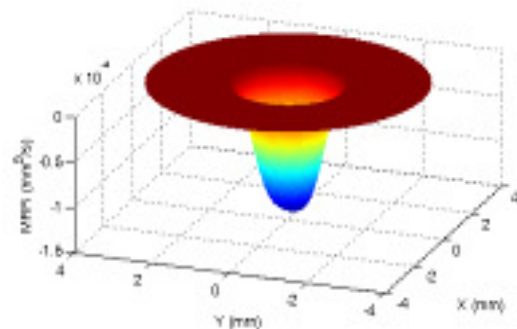


Figure 6 Simulated Tool Influence Function

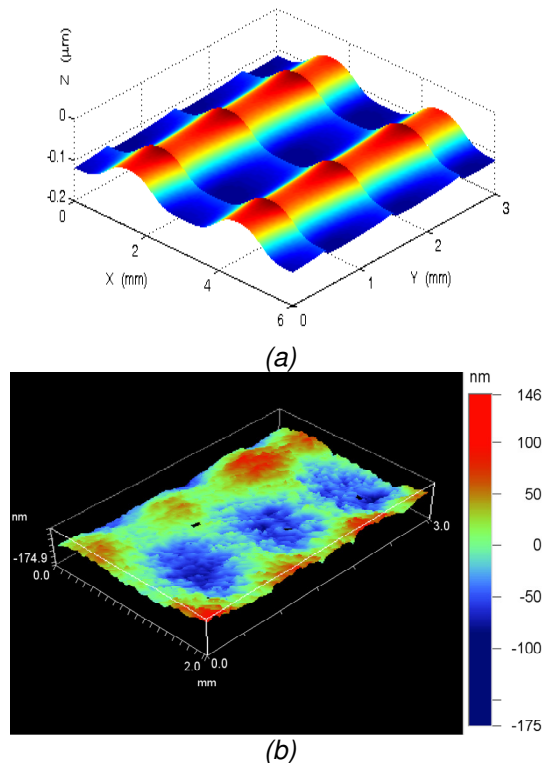


Figure 7. (a) Simulated and (b) measured surface topographies of the polished structured surfaces.

CONCLUSION

3D-structured surfaces have been widely used in different applications. Computer Controlled Ultra-precision Polishing (CCUP) is an enabling processing technology which is capable of fabricating ultra-precision freeform surfaces with sub-micrometer form accuracy and surface roughness in the nanometer range. However, the generation of 3D structured surfaces by CCUP still depends largely on the expensive trial-and-error approach when new materials, new designs of the 3D-structured surfaces or new machine tools are used. There is a need for the study of modeling methods which can simulate and predict the surface generation during CCUP of 3D structured surfaces.

This paper presents a surface topography simulation model for theoretical modelling and simulation of computer controlled ultra-precision polishing of structured surfaces. The model has been preliminary evaluated through a series of simulation and polishing experiments. The prediction results are found to agree well with the experimental results.

ACKNOWLEDGEMENT

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Investigation of the Dynamic Grinding Forces Resulting from the Machine Compliance Leading to Regenerative Chatter

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Introduction

In precision grinding the depth of cut is so small that even slight vibration can have dramatically damaging effects on surface finish, wheel wear, and form accuracy [1, 2, 3].

As the grinding wheel contacts the workpiece, material is removed and the wheel begins to wear. The complex interaction in the zone of contact depends on the dynamic compliance of the grinding machine, contact compliance of the grinding wheel and the dynamic cutting forces. Vibration causes dynamic deformation of wheel, spindle, and machine leading to unwanted, visible patterns on the workpiece.

It is important to reduce the magnitude of the time-varying component of grinding forces with adequate choices of process parameters, i.e. peripheral speeds, feeds, depths of cut, along with proper wheel truing and dressing. However, because of the numerous parameters involved in the grinding process and of the rather subtle interactions taking place between them, it is difficult to distinguish cause and effect.

Custom Machine Tool

Grinding machine configurations, such as ID or OD grinding are difficult to instrument because of the rotating bodies of both the workpiece and abrasive wheel. Vibration studies, whether regenerative or forced, are difficult to explore on most commercially-available grinding machines. Therefore, the motivation for developing and building custom-instrumented air bearing spindles [Professional Instruments] can meet many of the challenges for precision grinding in this work.

The instrumentation in this work addresses the need for precision grinding feedback through the use of a spindle with embedded non-contact displacement sensors that measure the relative axial motion between the rotor and stator in addition to non-contact displacement sensors to

measure the commanded table position and the workpiece's material removed.

Our experiments are performed on an ultra-precision lathe [Nanotechnology Systems]. The linear axes are fully-constrained, oil hydrostatic box way slides with linear motors. The system is suitable for both single point diamond turning and deterministic micro-grinding of optical components and has successfully turned a mirror with surface roughness below 1.6 nanometers Ra.

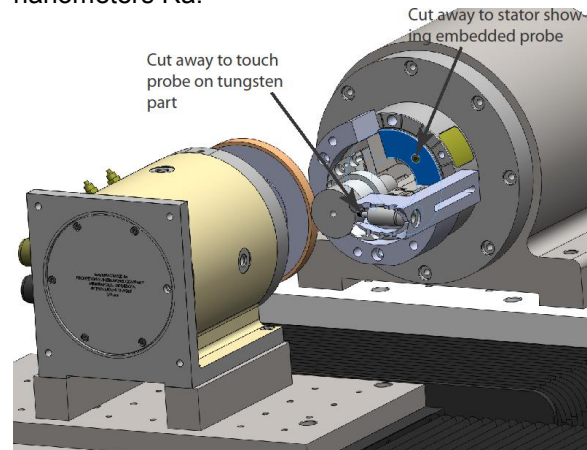


FIGURE 1. Setup to measure cutting forces and material removed from the part.

In this work, we provide new understanding of the interrelationships between grinding parameters and the visible component of surface finish.

Machine Compliance

The machine tool's structural loop compliance acted upon by the grinding forces cause deflections which are variable and large in comparison to the accuracies required. The deflection of the machining system causes

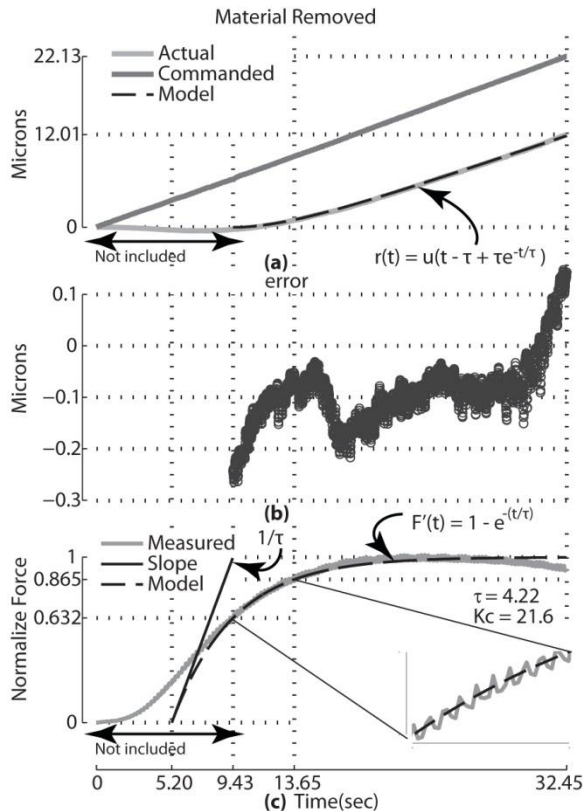


FIGURE 2. Modeling of the relationship between the programmed in-feed rate and the normal grinding force as a first order system

the actual position of the wheel-workpiece interface to lag behind the machine's commanded position as shown in Figure 2.a.

Typically this response has been modeled as a first order system characterized by a time constant, τ , and the differential equation.

$$\tau \frac{dv}{dt} + v = u$$

$$v = r(t)$$

Where $r(t)$, is the radial reductions of the workpiece.

The time constant reflects the response speed of the grinding control system and in control engineering terms, the value of the time constant equals the time lag or delay during the steady state period of a ramp type input signal. However, as seen in Figure 2.a there is a considerable difference between the ramp of the commanded in-feed and the actual in-feed that an offset was required in the model.

For this work the time constant is related to factors such as grinding system stiffness, the grinding wheel, and the workpiece material.

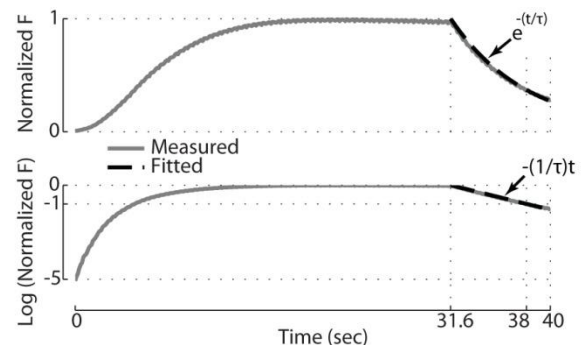


FIGURE 3. Calculation of the grinding system time constant during dwell period.

There are a number of ways to calculate the grinding system time constant. One method takes the log of the normalized force during a dwell cycle as shown in Figure 3. The time constant is the inverse of the negative slope to the log dwell.

Using the dwell period method means optimizing the feed cycle using the value of the time constant computed from previous grinding operations rather than from the operation which is currently being undertaken. Also the time constant may change not only between consecutive cycles but between the in-feed and the dwell cycle for spark-out. That was witnessed first-hand in this work.

Another method uses the in-feed cycle as shown in Figure 2.c. Normalizing the measured force resembles a unit step response of a first order system. One important characteristic of the normalized exponential response curve is that at $t = \tau$ the response has reached 63.2% of its total change. For in-process radial control the force signal can be differentiated and the inverse slope to the tangent line represents the time constant of the system.

In this work accurate estimation of the radial material removal was highest when determining the time constant during the initial in-feed. The errors between the commanded and actual in-feed were less than half a micron as shown in Figure 2.b.

Comparing Figure 2.a and 2.c it is shown that the grinding force and the workpiece radius

reduction did not start to increase at the same time. The grinding force starts to increase when the wheel comes in contact with the workpiece, but the workpiece radius reduction increases behind the force. This phenomenon has been previously observed and believed to be a result of friction between the wheel and workpiece rather than the material removal process [4].

For this work the delay between the material removal process and friction was equal to the time it took the measured force to reach 63.2% of its total change. Using this delay in the model corresponded well with the measured data as shown in Figure 4 and Figure 5 for a variety of parameters which yielded errors below half a micron in size for the work.

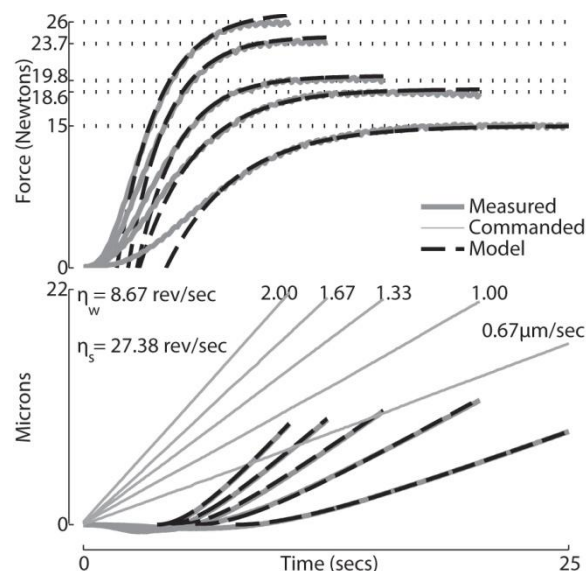


FIGURE 4. First order model with delay compared to measured data with different in-feeds. The work was a 2" diameter tungsten carbide puck spun to 8.67rev/sec and the wheel was a 7" diamond 320grit resin bonded spun to 27.38rev/sec.

Dynamics

The first order model approximation assumes a linear cutting force, $F = K_c a$, in which the normal grinding force is equal to the product of a cutting stiffness parameter, K_c , and the instantaneous cutting depth, a . Also the force is assumed to reach a steady state value affecting only the final size of the part. Both, however, are not so. From the measured data in Figure 2.c the cutting force is not linear and fluctuates around an average value. This fluctuation results in varying deflections between the wheel and

workpiece which affects the form and visible finish of the part.

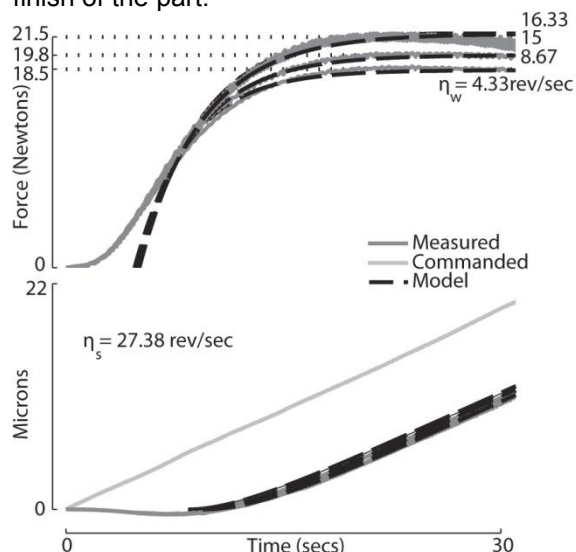


FIGURE 5. First order model with delay compared to measured data with different work speeds. The work was a 2" diameter tungsten carbide puck and the wheel was a 7" diamond 320grit resin bonded spun to 27.38rev/sec. The in-feed was a constant 0.67microns/sec.

In Figure 2.c the normalized grinding force data has been average over the circumference of the part so that there is one sample per revolution of the part. In Figure 6 the un-averaged data, both in time and around the circumference of the work is shown. Here the self-excited vibration occurs as a consequence of machine vibration interacting with the depth of cut also known as regenerative chatter.

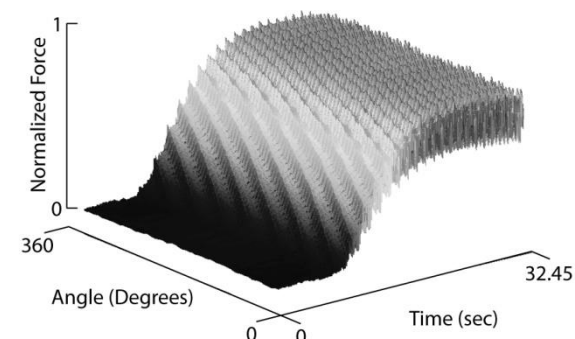


FIGURE 6. Measured grinding force from Figure 2.c re-shaped to show the force around the circumference of the tungsten carbide puck.

Chatter in any cutting process arises due to the interaction between the dynamic characteristics of the machine tool structure and those of the

cutting process. Figure 7 shows the measured receptance of the machine tool structure. The low frequency resonance is believed to contribute to the chatter in this work.

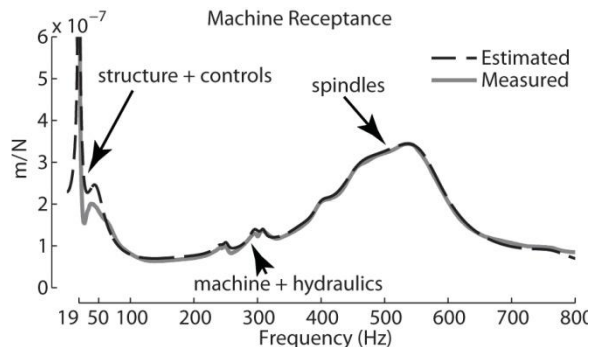


FIGURE 7. Measured and estimated grinding machine receptance.

The estimated receptance in Figure 7 can be used in a model similar to the one developed by Snoeys and Brown [5]. An analogue model which included the possibility of regeneration on both the grinding wheel and workpiece, and the effects of the wheel flexibility, which was assumed to be a contact stiffness proportional to the wheel width. Using this model and empirical results from measured data the regenerative chatter may be estimated and avoided.

The model does not account for force vibrations. In Figure 8 the wheel is now traverse at 55mm/min across the part instead of a plunge grinding operation. The wheel was spun at 45.17rev/sec while the work is spun at (a) 6.18rev/sec, (b) 6.45 rev/sec, and (c) 6.72rev/sec. The measured force shows that an integer speed ratio of 7, as in case (b), caused the imperfect form on the workpiece imparted by the imbalanced wheel to catch the crest of the wave and repeat this imperfection into the workpiece as the wheel traversed across. As a result the part had a seven lobe form error. At non integer speeds such as in (a) and (c) this imperfect shape was obliterated. While speed ratios are important for reducing chatter the complete fix is to have a perfect wheel both trued and balanced.

It is apparent from this work that there are many significant parameters in grinding and that no single model explains all the observed phenomena. The unique instrumentation used here connects the theory with actual results to optimize the process and improve the quality of the workpiece.

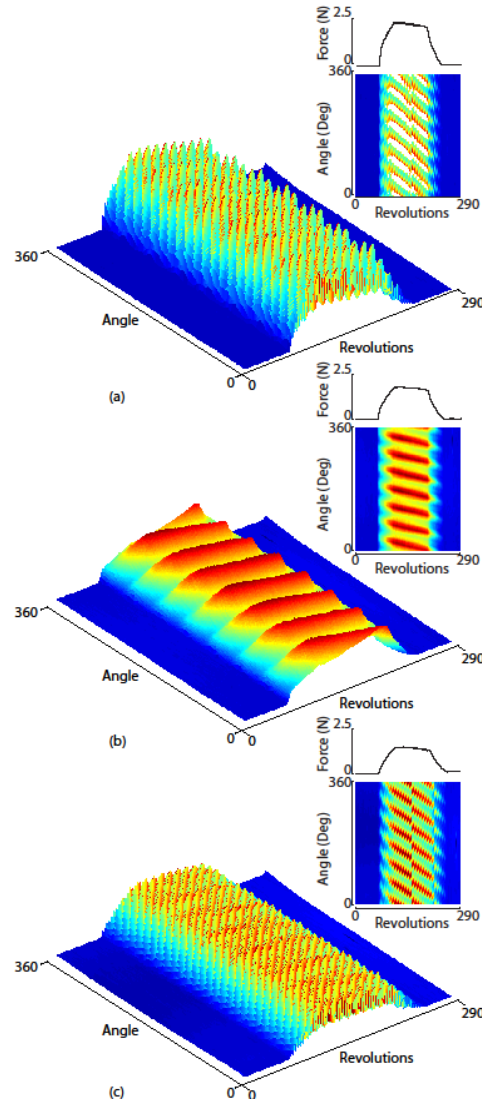


FIGURE 8. Effects of force vibration due to eccentric wheel with speed ratios.

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Ultra Precision Cutting of Tungsten Carbide Mold with PCD Milling Tool fabricated by EDM and Grinding

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INTRODUCTION

Recently, demands of green technology are increasing and the Fresnel lenses need to be installed to the solar panel as shown in Figure 1 in order to improve a generating efficiency. These lenses must be made of glass to endure high temperature and must be low in cost. Consequently, these must be press-molded in high temperature by using precision ceramic molds such as tungsten carbide or SiC. These ceramic molds are usually ground by using the diamond wheel [1]. The diamond wheel must be trued on the machine before grinding. However, it takes a long time to true and dress the grinding wheel, and the grinding wheel will wear, and it is difficult to maintain the original geometrical shape of the wheel [2].

In this study, large milling tool made of poly-crystal diamond (PCD) having excellent wear resistance is developed in order to generate the precision Fresnel shape mold made of tungsten carbide. The tool edge is machined as a sharp knife-edge with wire electric discharge machining (EDM), and 80 cutting edges are ground on the tip of the tool with the diamond wheel. In the cutting test, the Fresnel shape mold made of binder-less tungsten carbide was cut with the developed PCD milling tool by the developed simultaneous 2-axis (Y, Z) controlled cutting method.

PCD MILLING TOOL

The PCD milling tool was fabricated as shown in Figure 2. At first the PCD wafer was bonded with a silver alloy to a cemented carbide shank and the side surface of the PCD wafer was cut into a conical shape by wire EDM. Finally, the cutting edges were ground with a sharp diamond wheel. Figure 3 shows photographs of a newly developed poly-crystal diamond (PCD) milling tool. In addition, Specifications of the PCD tool are shown in Table 1. The grain size of the PCD was 5 μm . The outer diameter was $\Phi 24$ mm and 80 cutting edges were ground on the tool edge.

The land width on the cutting edges was 0.03mm.

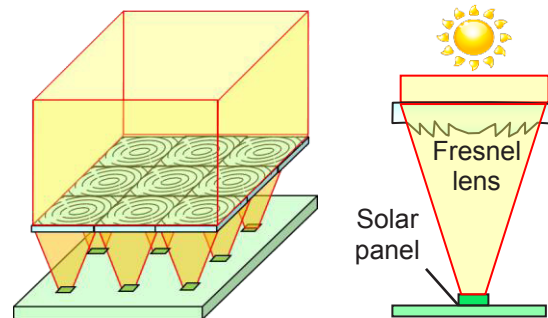


FIGURE 1. Fresnel lens for solar panel

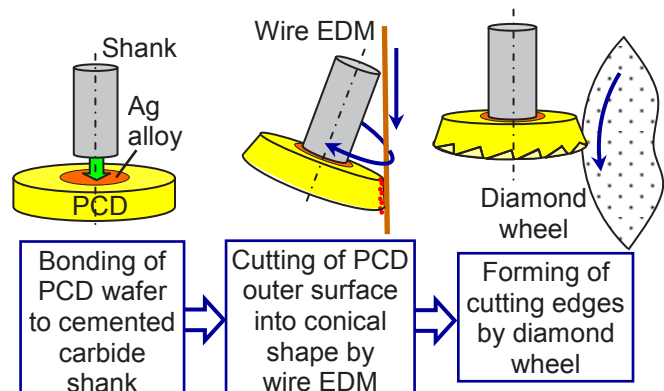
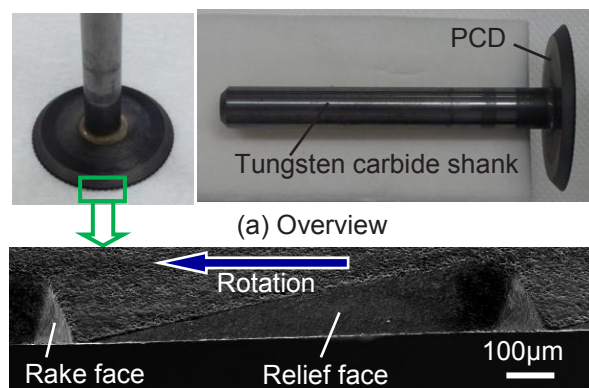


FIGURE 2. Fabrication process of PCD milling tool



(b) SEM image of cutting edges
FIGURE 3. Developed PCD milling tool

TABLE 1. Specifications of PCD milling tool

Tool material	Poly-crystal diamond
Particle size	5 μm
Diameter	$\Phi 24$ mm
Number of cutting edge	80
Rake angle	-20 degs.
Land width	0.03 mm

EXPERIMENTAL SET-UP AND METHOD

The tool was chucked to an air-bearing spindle with a collet chuck on Y-axis table and the spindle was attached to a simultaneous 4-axis (X, Y, Z, C) controlled ultra-precision machine, ULG-100D(SH3) as shown in Figure 4. A maximum rotational speed of the tool air-bearing spindle is $40,000 \text{ min}^{-1}$. The tool was activated on X, Y and Z-axis by the linear scale feedback system with 1 nm positioning resolution. The structured workpiece was vacuum-chucked onto a workpiece air spindle (C-axis table). A maximum rotational speed of the workpiece air spindle is $1,500 \text{ min}^{-1}$. As a coolant, the oil mist was used. And, a non-contact laser probe scanner, NH-3UP and a non-contact measuring instrument, ZYGO New view 6200 were used in order to measure the machined workpiece. The NH-3UP has blue laser of short wavelength ($\lambda=0.473 \mu\text{m}$) as a laser probe in order to reduce a laser spot diameter and improve a resolution [3]. Then, it can be measure the structured shape precisely.

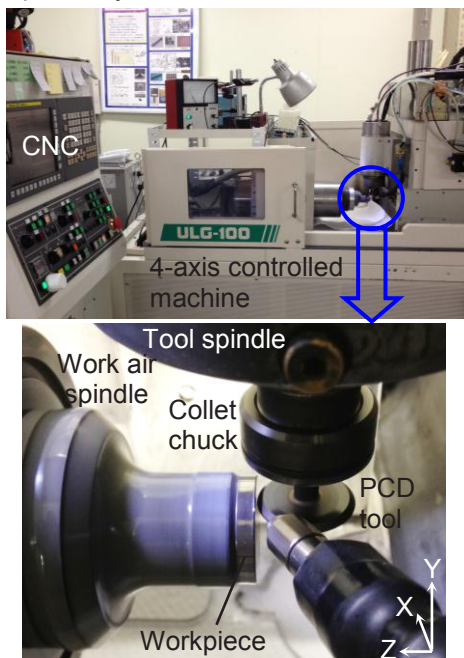


FIGURE 4. View of cutting experiments

BASIC CUTTING PERFORMANCE

In order to evaluate a basic cutting performance, flat shape of tungsten carbide was cut with the developed PCD milling tool as shown in Figure 5. Table 2 shows cutting conditions. The feed rate and the depth of cut were changed. Figure 6 shows the effect of the feed rate on the area of released tungsten carbide particle and surface roughness. Figure 7 shows the effect of the depth of cut on the area of released tungsten carbide particle and surface roughness. These data were measured by the non-contact measuring instrument. As the feed rate increases, the surface quality decreases by the dropout of tungsten carbide particle occurs easily. However, in case of small range of the feed rate and the depth of cut, mirror surface of machined tungsten carbide was obtained.

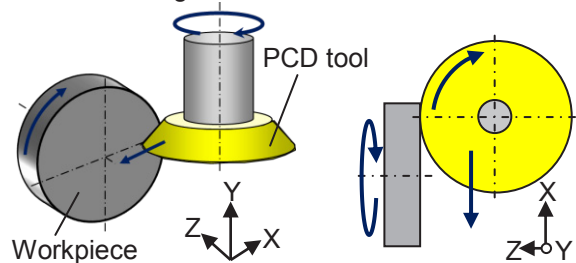


FIGURE 5. Flat cutting of tungsten carbide

TABLE 2. Cutting conditions

Tool material	Poly-crystal diamond
Particle size	5 μm
Rotational rate	$18,000 \text{ min}^{-1}$
Workpiece	Tungsten carbide
Rotational rate	500 min^{-1}
Feed rate	0.5, 1, 2, 3, 5 mm/min
Depth of cut	0.5, 1, 2, 3, 5 $\mu\text{m/pass}$
Coolant	Oil mist

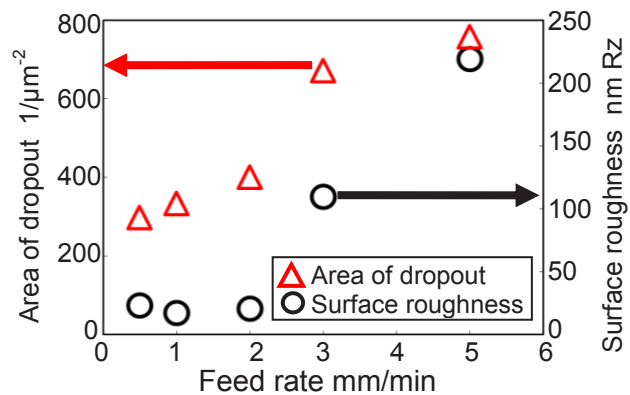


FIGURE 6. Effect of feed rate on area of dropout and surface roughness

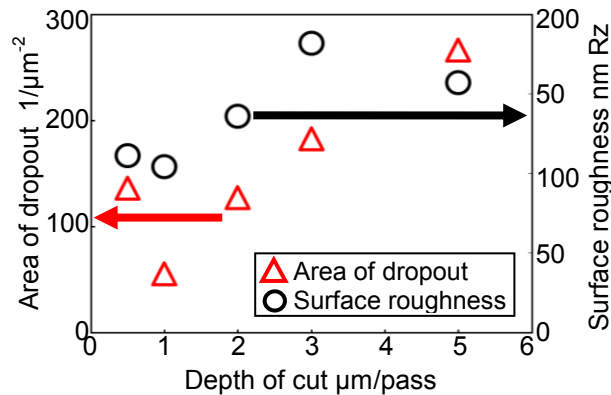


FIGURE 7. Effect of depth of cut on area of dropout and surface roughness

PRECISION CUTTING OF FRESNEL SHAPE

In this study, the cutting method with simultaneous 2-axis (Y, Z) controlled cutting for generating the Fresnel shape was developed as shown in Figure 8. The PCD milling tool scans vertically along the workpiece radial position from center position to outer position. The feature of this proposed method is that the tool rotates in the parallel with the workpiece rotational direction at the cutting point. The axis-symmetric Fresnel shape of the mold will be cut by the PCD milling tool with sharp edge. Therefore the sharpness of the tool is important in order to machine the structured surface precisely.

The tool path is calculated by the following formulae:

$$\left. \begin{aligned} Y_0 &= Y_w - R \sin \theta \\ Z_0 &= Z_w - R(\cos \theta - 1) \end{aligned} \right\} \quad (1)$$

Where R is a radius curvature of the PCD tool tip, θ is a tangential angle of the cutting point, (Y_0, Z_0) is a center position of the PCD tool tip and (Y_w, Z_w) is a cutting point on the workpiece surface.

The axis-symmetric Fresnel shape (Figure 9) is given by:

$$Z = f(Y) = \text{mod}(g(Y), b) \quad (2)$$

$$g(Y) = \frac{C_v \cdot Y^2}{1 + \sqrt{1 - (K+1) \cdot C_v^2 \cdot Y^2}} + \sum_{i=1}^n C_i \cdot Y^i \quad (3)$$

Where Y is a radial position of the mold, Z is a rotation axis position, K , C_v , C_i ($i=1 - n$) are aspheric constants, and b is a height of Fresnel lens.

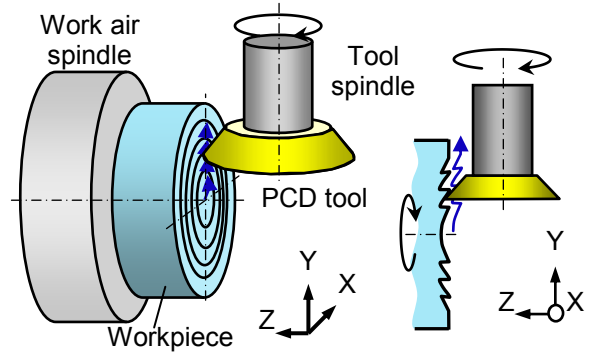


FIGURE 8. Developed cutting method of Fresnel shape

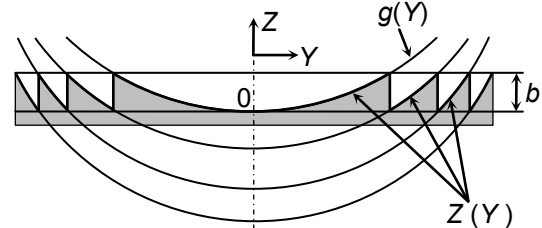


FIGURE 9. Definitions of Fresnel shape

EXPERIMENTAL RESULTS OF MACHINED FRESNEL MOLD

In order to evaluate the cutting performances for the Fresnel shape, a tungsten carbide workpiece of 28 mm diameter was cut into the shape as shown in Figure 10. Cutting conditions are shown in Table 3. Figure 11 shows a measured profile of the Fresnel mold which was measured by the non-contact laser probe scanner. It was found that structured surface with edges was cut sharply. Figure 12 shows a deviation profile of the Fresnel mold which was calculated from the measured profile of Figure 11. The form accuracy was less than 0.4 μm P-V. Figure 13 shows a surface roughness profile of the machined surface which was measured in the center region by the non-contact measuring instrument. The surface roughness of 25 nm Rz was obtained. However, some tungsten carbide particles on the machined surface were dropped out and a part of Fresnel edge was chipped as shown in Figure 14.

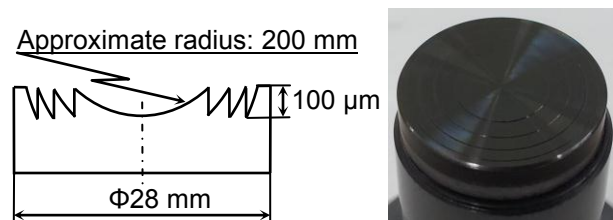


FIGURE 10. Geometry and view of Fresnel lens mold

TABLE 3. Cutting conditions

Tool material	Poly-crystal diamond
Particle size	5 μm
Rotational rate	18,000 min^{-1}
Workpiece	Tungsten carbide
Rotational rate	500 min^{-1}
Depth of cut	0.5 $\mu\text{m}/\text{pass}$
Feed rate	0.05 mm/min
Coolant	Oil mist

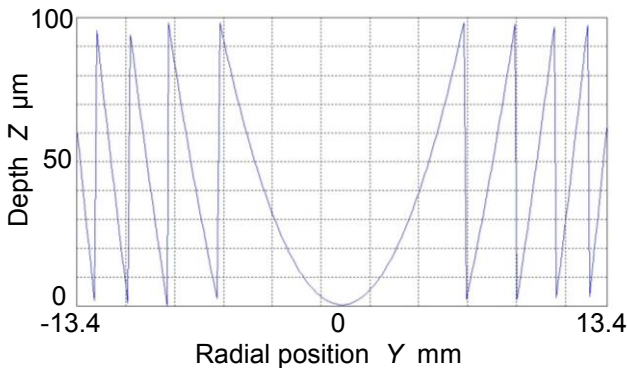


FIGURE 11. Measured profile of Fresnel mold

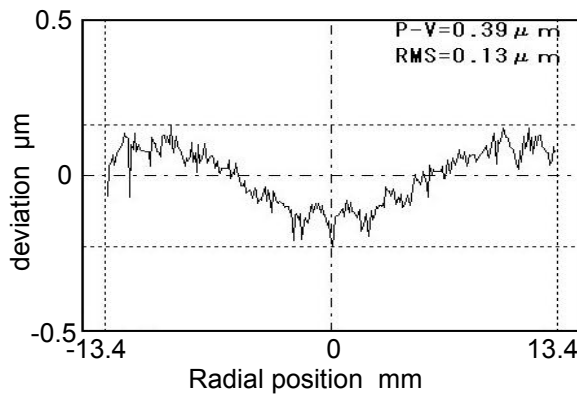


FIGURE 12. Form deviation profile of Fresnel mold

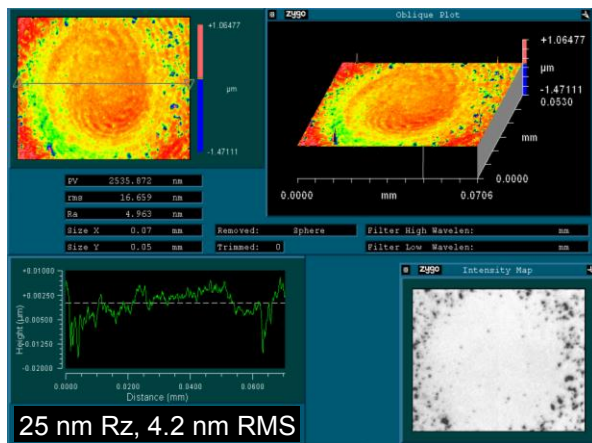
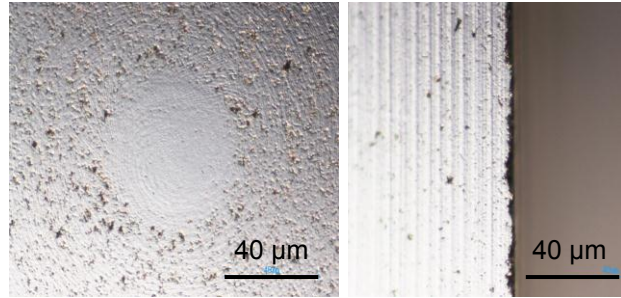


FIGURE 13. Surface roughness profile of machined Fresnel mold



(a) Center region

(b) Fresnel edge

FIGURE 14. Nomarski photographs of Fresnel mold

CONCLUSIONS

The PCD milling tool was proposed and developed in order to machine the precision Fresnel mold for the solar panel. The PCD tool was fabricated by wire EDM and grinding and the tungsten carbide mold was cut with the developed PCD tool. From the experiments, the following results were obtained:

- (1) The PCD milling tool could be fabricated by wire EDM and grinding and the cutting edges were generated sharply.
- (2) The Fresnel lens mold was cut precisely with the developed PCD milling tool, the form accuracy of 0.4 μm P-V and the surface roughness of 25 nm R_z were obtained. It was found that the developed PCD milling tool fabricated by wire EDM is effective for generating the Fresnel mold.

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DEVELOPING CAPABILITIES FOR FABRICATING EXTENDED POLYNOMIAL FREEFORM MIRRORS

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INTRODUCTION

An optical system has been designed at Sandia National Laboratories that includes an extended polynomial freeform mirror. The envisioned mirror is described by a polynomial series in X and Y such that the best fabrication method is direct machining via slow tool servoing where the desired surface geometry is generated by synchronizing linear tool position (X, Y, Z) with spindle position (theta). The fabrication and metrology of freeform surfaces involves complex processes with numerous inputs and potential error sources. Therefore, standard axis-symmetric surface geometries were fabricated in a nonstandard, non-rotationally symmetric form in order to generate and demonstrate fabrication and metrology processes for more arbitrary freeform geometries. Prior research in fast tool servo¹ and slow tool servo^{2,3} equipment development have relied on tilted flats and off-axis parabolas since they facilitate the machining of non-rotationally symmetric surfaces that can be oriented to leverage on-axis metrology techniques. Such complexity in fabrication and simplicity in metrology prompted their use in the recent work.

OFF-AXIS OPTIC SURFACE REQUIREMENTS

The desired extended polynomial freeform can be described by the equation

$$z = \frac{R_c r^2}{1 + \sqrt{1 - (1 + k)R_c^2 r^2}} + \sum_{i=1}^N A_i E_i(x, y)$$

where z is the surface sag at position x, y , r from the optic center apex, k is the conic coefficient, R_c is the radius of curvature, A_i is the coefficient on the i^{th} extended polynomial term, N is the number of polynomial terms in the series, and $E_i(x, y)$ represents the i^{th} order power series term in x and y . The final design prescription for the desired optical surface was based on a 5th order polynomial where optical system requirements

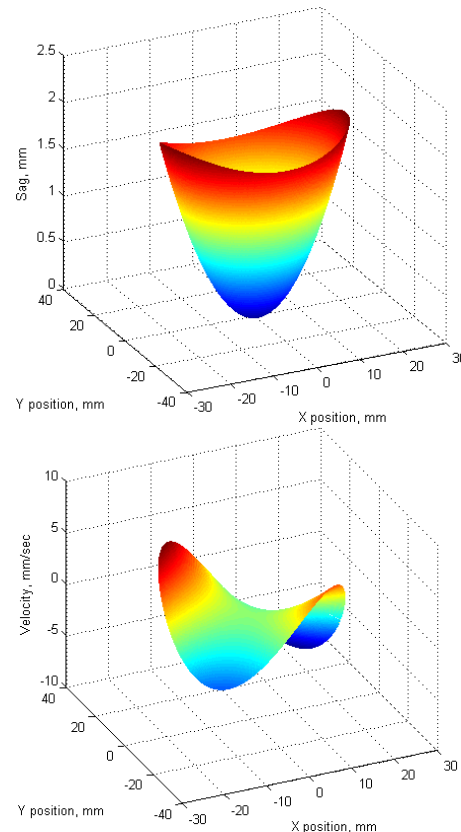


FIGURE 1. The extended polynomial freeform surface and its z-axis velocity profile.

necessitate a surface form accuracy better than 1 μm and a surface finish better than 5 nm Sa. Figure 1 shows the desired surface geometry and resultant z-axis velocity profile for slow tool servoing. The optic has an aperture diameter of 41.35 mm, a sag of 2.02 mm, and a maximum rotational asymmetry of 417 μm at its aperture diameter. The maximum z-axis tool velocity required during cutting is 6.46 mm/sec. While the tilted flat and off-axis parabola cut during process development did not replicate the same asymmetrical departure as that of the final freeform design, they were representative of early design revisions and were found adequate

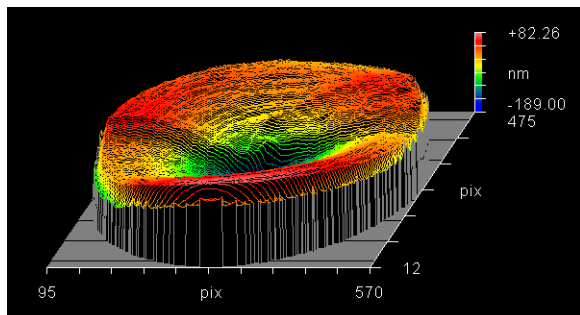


FIGURE 2. 0.271 μm peak-to-valley form error for the tilted flat.

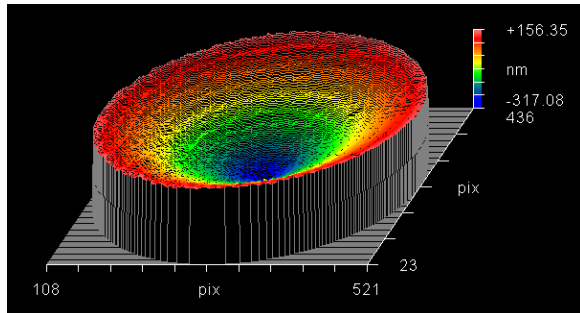


FIGURE 3. 0.473 μm peak-to-valley form error for the face cut flat.

for examining the form accuracy and surface finish that could be achieved through slow tool servoing. Cutting experiments were all performed on 5 mm thick, $\frac{1}{2}$ hard, alloy 464 naval brass due to its low tool wear and excellent surface finish under diamond turning. Rough cutting on each of the test workpieces was performed using a 0.244 mm radius diamond tool with the final surface being machined using a single finish pass with a 0.38 mm radius diamond tool.

TILTED FLAT FABRICATION

Process work began with the tilted flat as it was the simplest geometry to program, to fabricate via slow tool servoing and to measure in a standard manner via on-axis form interferometry. The initial concern raised in process evaluation was whether the cutting speeds required for slow tool servoing would generate surface finishes that met system requirements. Since slow tool servoing requires coordinated motion of the z-axis with the x-axis and the work spindle, cutting velocities decrease with increases in the asymmetric departure and can therefore be more than an order of magnitude lower than that utilized in traditional turning operations. Subsequently, questions were raised regarding the potential for degradations in the material removal process (ex. tearing or burr formation), in the behavior of

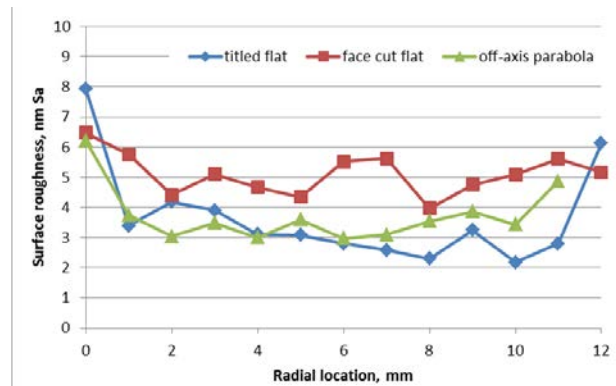


FIGURE 4. Surface finish versus radial position for each test workpieces.

the cut material (ex. chip buildup or scratching) or in the rate of tool wear.

A 25.4 mm tilted flat was cut with 100 μm of rotational asymmetry at its OD which required a maximum z-axis velocity of 0.84 mm/sec. For comparison, a 25.4 mm flat was machined via traditional face turning under typical process feeds and speeds. While the cutting velocity for the tilted flat varied with both x-axis and spindle position, its maximum spindle speed was on the order of 170 rpm with a finish cross feed rate of 1 $\mu\text{m}/\text{rev}$ and a depth of cut of 10 μm . The face cut flat, however, was finish machined at 2000 rpm using a cross feed rate of 1.5 $\mu\text{m}/\text{rev}$ and a depth of cut of 3 μm . Figures 2 and 3 show on-axis interferometric measurements of the tilted and face cut flats, respectively. Unexpectedly, the 0.271 μm form error of the tilted flat was almost half the 0.473 μm form error of the face cut flat. While the error for the face cut flat was not examined in great detail, its most likely source is believed to be deformation due to vacuum mounting. Figure 4 plots surface finish across the radius for the flats and the off-axis parabola. No degradation was observed during slow tool servoing as the surface finish for the face cut flat was generally rougher than that of the tilted flat, except at part center where both cutting processes produce a cutting velocity that approaches zero. Both flats exhibited roughness values, however, near or below the system requirement of 1 nm Sa. While a secondary polishing step is unnecessary at this time, system characterization will evaluate whether further improvements are necessary.

OFF-AXIS PARABOLA FABRICATION

After examining the machining of the tilted flat, an off-axis parabola was fabricated to examine a

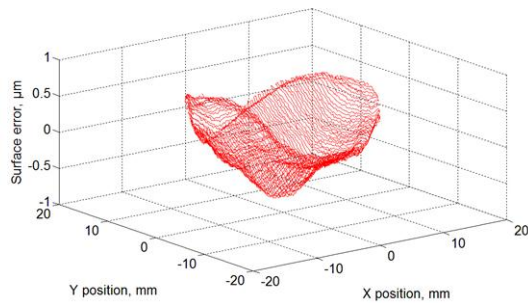


FIGURE 5. $0.96\ \mu\text{m}$ peak-to-valley form error for the off-axis parabola.

surface that more closely represented the desired freeform geometry. While a parabolic section can be cut in a rotationally symmetric manner, the part was defined in an off-axis frame, rotated about its center surface normal. Measurement and testing of the resulting parabolic mirror are straightforward since a parabola is well understood both geometrically and optically. As with the tilted flat, the parabola was fabricated while the optical design was still changing; and therefore, did not represent the final surface as originally intended. The off-axis parabola used a 6th order extended polynomial series centered in the off-axis pupil such that focus was 20 mm off axis with a tilt of roughly 17.5° . It had a 25.4 mm outside diameter, a sag of 1.18 mm and a maximum asymmetric departure of $176\ \mu\text{m}$. Its maximum z-axis travel velocity during cutting was 2.53 mm/sec.

The part was cut via slow tool servoing using a finish cross feed of $1\ \mu\text{m}/\text{rev}$, a depth of cut of $5\ \mu\text{m}$, and a maximum spindle speed of approximately 170 rpm. Figure 5 shows the $0.96\ \mu\text{m}$ surface form error obtained on Sandia's F-25 micro-CMM. While the measurement uncertainty for the F25 is roughly 250 nm; workpiece deformation under vacuum mounting, diamond tool offset errors, motion errors during slow tool servoing and micro-CMM probe offset errors are other sources for the observed form error whose individual contributions were not quantified. Work was performed to quantify potential errors between the optical model and the 3D solid models utilized during metrology. Analysis showed model correlation to better than 70 nm, significantly less than the observed errors. While the form error did increase over both the tilted flat and the face cut flat, it remains within the requirements for the optical system. As with the tilted flat, Figure 4 shows that the finish across the part radius remains at or below the 5 nm Sa

finish requirement. The part has also been tested optically using a HeNe laser to see that it properly focuses a collimated input. While focal distance has not yet been measured the focused spot diameter is roughly $10\ \mu\text{m}$, suggestive of diffraction limited performance. Interferometric data has also not been collected, but is planned in continued work.

CONCLUSIONS

Off-axis optics have been utilized to develop and evaluate a slow tool servo fabrication process for an extended polynomial freeform mirror. Surface requirements of $1\ \mu\text{m}$ for form accuracy and 5 nm Sa for surface finish have been demonstrated on representative tilted flat and off-axis parabola surfaces. Remaining work will fabricate and measure the desired freeform mirror for implementation in a first prototype system.

ACKNOWLEDGEMENTS

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STUDY ON MACHINING HIGH ASPECT RATIO MICRO-CHANNEL STRUCTURES VIA A SHAPING METHOD USING A DIAMOND TOOL

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INTRODUCTIONS

Recently, microstructures have been widely used in high-tech industries and applications including in the display, bio, and energy fields. Within the various microstructures, micro channel have been applied in security films, optical wave guides, micro heat sinks and bipolar plates of fuel cells. In order to increase the efficiency and performance of these products, high aspect ratio micro-channel structures are required [1][2][3][4]. The chemical and optical processing methods that have been used to machine micro-channel structures have primarily been machining methods; however, mechanical methods are preferable in the fabrication of micro-channel structures because they enable higher quality micro-channel structures to be fabricated. Micro-channel structures with increased aspect ratios (ARs) are easily deformed and destroyed via mechanical machining due to the required cutting force, as shown in Figure 1. Thus, methods of machining high aspect ratio channel structures without deformation and breakage need to be studied. The deformation and breakage of the micro-channel structures were analyzed according to machining processes, overlapped cutting depths and feed rates by machining micro-channel structures using shaper and then high aspect ratio micro channel structures with AR30 were machined based on earlier experiments in this study.

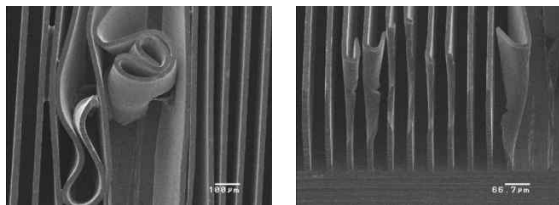
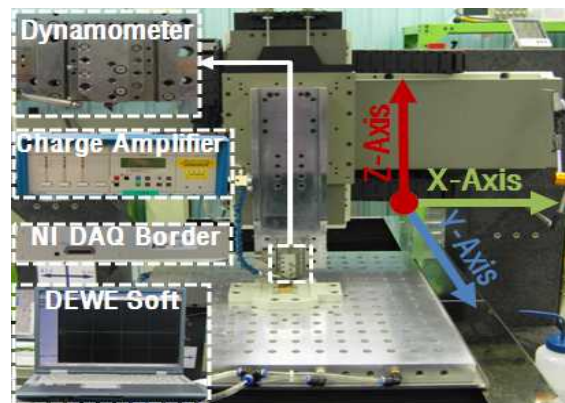


FIGURE 1. Deformation and breakage of micro-channel structures with high aspect ratio

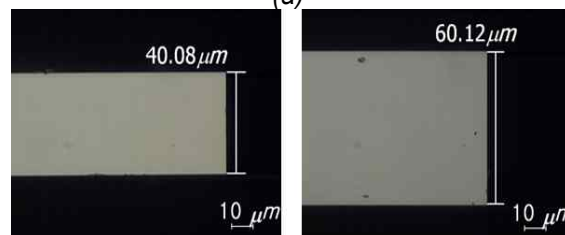
EXPERIMENTAL SYSTEM AND METHOD

The machining system used in this experiment with the shaper is presented in Figure 2(a). The system can control four-axis movement with 5 nm accuracy. The stroke of X, Y and Z is 900×900×100mm. The cutting forces of X, Y and Z-axes were measured using a tool dynamometer.

The rectangular diamond tool (width 40μm and 60μm, valid cutting length 400μm and 600μm, shape angle 0°, front clearance angle 3° and side clearance angle 1°) was designed and fabricated as shown in Figure 2(b).



(a)



(b)

FIGURE 2. (a) Schematic diagram of the experimental system and (b) the manufactured rectangular diamond tools

First, three machining methods were considered in order to analyze the effects of the machining processes on the deformation of the micro-channel structures (width $15\mu\text{m}$, depth $400\mu\text{m}$ and AR 27). In the first method, as shown in Figure 3 (a-I), the entire channel is machined in the Z direction with an increasing cutting depth and the micro-channel structures are completed with the final cutting depth. This method is the conventionally used method. In the second method, as shown in Figure 3 (a-II), the channels are machined individually. With the third method, as shown in Figure 3 (a-III), the micro-channels are machined with a double pitch and the middle of remaining channel is then machined further. Second, the micro-channel structure of AR30 (width $20\mu\text{m}$, depth $600\mu\text{m}$) was machined in order to analyze the effects of the overlapped cutting depths ($5\mu\text{m}$ and $10\mu\text{m}$) and feed rates (50mm/s and 100mm/s) on the deformation and breakage of the micro-channel structure as shown in Figure 3(b). The detailed experimental conditions are presented in Table 1.

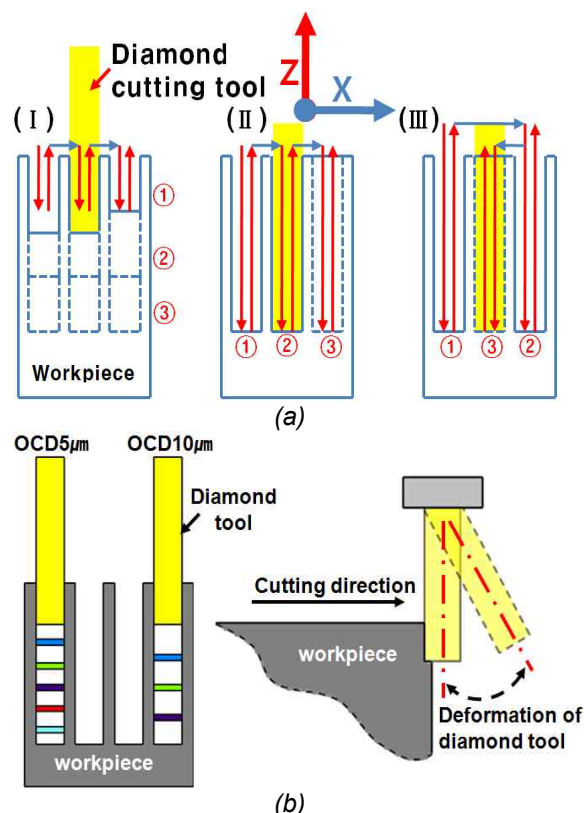


FIGURE 3. Experimental methods according to (a) the machining processes and (b) the overlapped cutting depths and feed rates

TABLE 1. Specifications of the experimental conditions

Workpiece	64 brass $30\times30\text{mm}^2$
Cutting tools	Diamond rectangular tools Width: $40\mu\text{m}$ and $60\mu\text{m}$ Lengths: $400\mu\text{m}$ and $600\mu\text{m}$
Feed rates	50 mm/s and 100mm/s
Pitch	$55\mu\text{m}$ and $80\mu\text{m}$
Cutting depth	Case1 ($2\mu\text{m}\times1$)+($5\mu\text{m}\times79$)+($3\mu\text{m}\times1$) Case2 Overlapped cutting depths $5\mu\text{m}$ and $10\mu\text{m}$
Cutting oil	Kerosene oil

EXPERIMENTAL RESULTS AND DISCUSSION

Figure 4 (a-I) presents the resulting micro-channel structures obtained using of the first method (machining the entire channel structures in the Z direction with increased cutting depth); Figure 4 (a-II) shows the results obtained using the second method (machining channel structures one by one; and Figure (a-III) presents the results obtained using the third method (machining channel structures with a double pitch and machining the middle of the remaining channels). Deformations were not seen in results of using the first method as shown Figure 4 (a-I) because each channel structure was machined from side to side without repeated loading as shown in Figure 4(b-I). The micro-channel structures were deformed in the opposite direction to the feed in the second method as shown in Figure 4 (a-II). Every second channel was deformed in two directions and the deformed channels prevented the deformation of other channels in the third method as shown in Figure 4 (a-III). With the second and third methods, each channel structure was continuously machined from a single side; as a result, the thin channel structures were subjected to repeated loading. Eventually the channel structures deformed in the direction opposite to the feed direction due to fatigue deformation, as shown in Figure 4 (b-II) and (b-III). Thus, it is desirable to use the first method to machine high aspect ratio micro-channel structures.

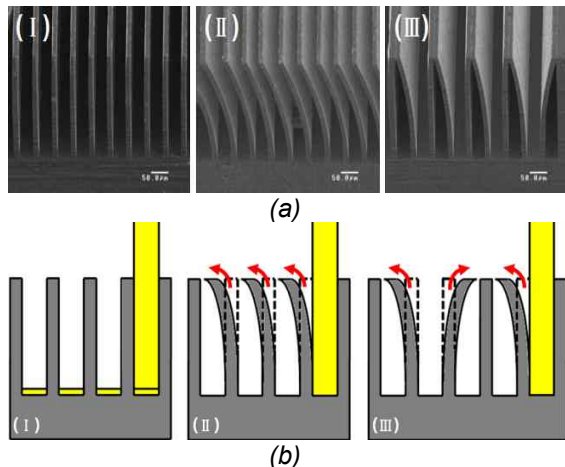


FIGURE 4. (a) Comparison of the deformation of the high aspect ratio micro-channel pattern structures according to the three machining methods and (b) an analysis on the deformation of the micro-channel structures according to the machining processes

The machined channel structures are comprised according to the overlapped cutting depths and feed rates as shown in Figure 5. There were no deformations and breakages of channel structures when using $5\mu\text{m}$ of overlapped cutting depth. In the application of $10\mu\text{m}$ of overlapped cutting depth, deformations and breakages of channel structures occurred as shown in Figure 5 because the initial entry shock of the tool and workpiece was increased more than that of the $5\mu\text{m}$ of overlap cutting depth. It was confirmed that the deformation and breakage of the micro-channel structure was much influenced by the overlapped cutting depth than the feed rate.

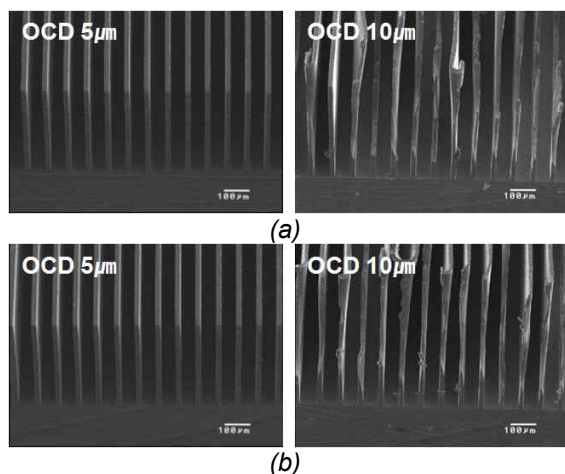


FIGURE 5. Comparison of the machined high aspect ratio micro-channel structures with overlapped cutting depths (OCD) of $5\mu\text{m}$ and $10\mu\text{m}$

overlapped cutting depths ($5\mu\text{m}$ and $10\mu\text{m}$) and feed rates of (a) 50mm/s and (b) 100mm/s

Based on earlier experimental results (a- I, $5\mu\text{m}$ of overlapped cutting depth and feed rate 100mm/s) micro-channel structures (width $20\mu\text{m}$, depth $600\mu\text{m}$, AR 30) with a high aspect ratio were machined without deformation or breakage as shown in Figure 6.

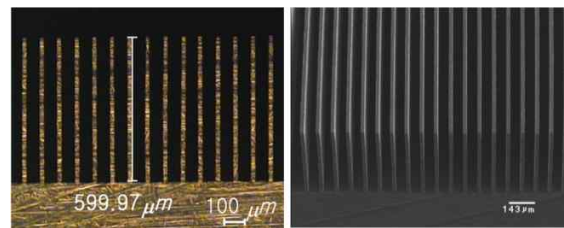


FIGURE 6. Machined micro-channel structures having high aspect ratio (width $20\mu\text{m}$, depth $600\mu\text{m}$ and AR 30)

CONCLUSIONS

The deformation and breakage of micro-channel structures were confirmed to result from the machining processes, overlapped cutting depths and feed rates through the fabrication of micro-channel structures using a shaper. High aspect ratio micro-channel structures of AR 30 were machined based on the earlier experiments in this study. The significant findings are as follows.

1. There was no deformation of micro-channel structures when using the first method, where the entire channel was machined in the Z direction by increasing the cutting depth but the micro-channel structures were deformed when using the second and third methods due to the fatigue deformation when the channels were continuously cut from a single side.
2. It was confirmed that the deformation and breakage of micro-channel structure is influenced more by the overlapped cutting depth than the feed rate.
3. It was confirmed that the first machining method and appropriate overlapped cutting depths are preferable for machining micro-channel structures with a high aspect ratio.

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REAL-TIME POSITION ERROR COMPENSATION FOR SLENDER WORKPIECE FOR LASER MACHINING

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INSTRUCTION

The demands for precision parts with micro features have been increasing in biomedical area. For example, catheters and stents are used for brain or heart surgeries and more and more they are required to have certain patterns on metal tubes for drug delivery or supporting blood vessels. Stents are only a few millimeters in diameter and machined by using laser and/or electrochemical machining (ECM) [1]. Because the workpieces in this biomedical application are generally long and slender, if the surface profiles of them are not straight, it is not easy to keep a laser beam focused on their surfaces even though laser machining itself is a non-contact machining process.

To prevent a laser beam from losing its focus while machining, active compensation for positioning error is desired. In that case, the surface profile should be measured in real-time and sent back to positioning stage for compensation. Since contact type measuring sensors cannot be used for a slender workpiece, two non-contact type sensors were tested and their experimental results are discussed in this paper.

COMPENSATION SYSTEM AND SENSORS

Figure 1 shows a concept picture of the active position error compensation system. A sensor block measures the position of a slender workpiece in XY-plane and its positioning error with respect to the laser beam is sent back to the workpiece feeding stage. While a XYZ stage keeps feeding the slender workpiece toward the focused laser spot, the positioning error induced by the barrel profile of the workpiece is compensated actively by the XYZ stage control system. There is another positioning XYZ stage in the right of the Figure 1. It is a manual stage to adjust the sensor block position with respect

to the laser beam and it does not move after the initial sensor block setting.

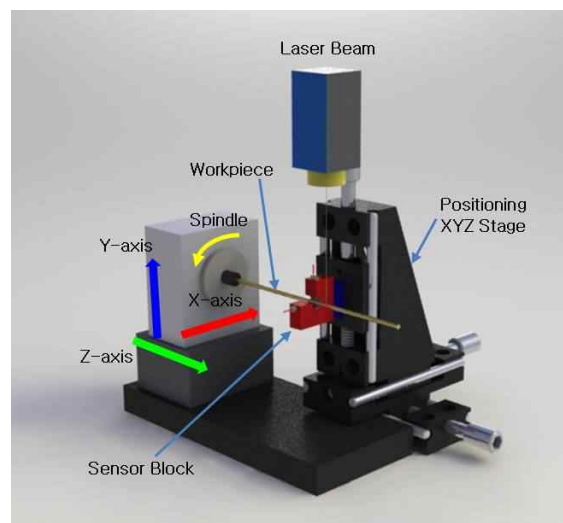


FIGURE 1. A conceptual configuration of an active position error compensation system for laser machining.

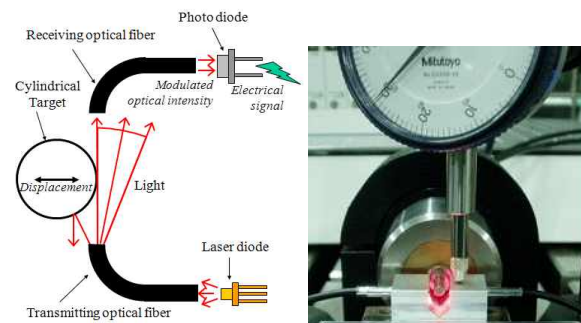


FIGURE 2. Principle of the edge detecting sensor and its feasibility test.

Two non-contact type sensors are tested in this research. The first sensor [2] uses the light

intensity passing over the edge as shown in Figure 2. Two optical fibers are facing each other, the one emits light and the other receives the incoming light and delivers it to a photo diode. The cylindrical object hides a part of light in the middle of the light path and changes the light intensity measured by the photo diode.

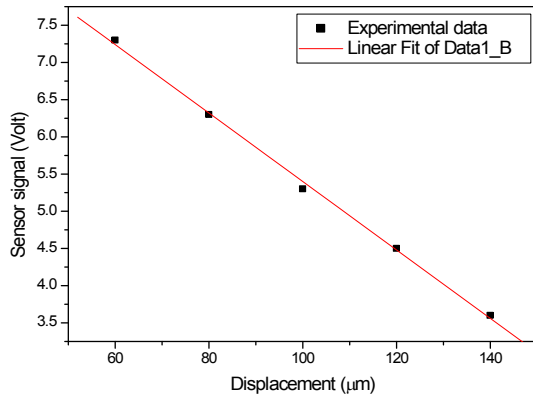


FIGURE 3. With a 3 mm diameter rod, the edge detecting sensor shows around 0.19 μm resolution considering its signal noise level.

From the feasibility test, it was found that this edge detecting sensor might have an advantage of measuring the position of very small diameter cylindrical rods, because it uses only shadows created by the edge of the rod. Other non-contact sensors such as capacitive sensors or eddy current sensors, require a certain target area for sensing and it is not easy to find these sensors for small diameter objects.

Although the sensor signal noise needed to be more suppressed to obtain better sensor resolution, the edge detecting sensor was applied to a real machining test as shown in Figure 4.

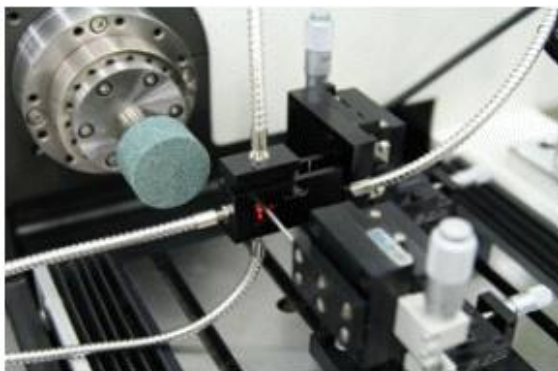


FIGURE 4. An XY edge detecting sensor was put under test for precision grinding application.

Since a laser module is complicated to be setup for a preliminary test, an XY edge detecting sensor was made and put in precision grinding process to compensate for the positioning error. The edge detecting sensor revealed a big drawback when it was applied to precision grinding test. It was a contamination problem from the machining process. The light intensity changed not only by the workpiece position but also by the particles and dusts from the grinding. If the edge detecting sensor is applied to laser machining, it is expected that similar problem may occur unless the machining spot is far away from the measuring point.

The second sensor considered to be used for positioning error compensation was a capacitive sensor. It has limitations on target size. The diameter of a cylindrical workpiece should be comparable to that of capacitive sensor. But its installation is relatively easy and contamination during machining is not a problem for this sensor.

EXPERIMENTAL SETUP

Figure 5 shows the experimental setup with capacitive sensors installed in XY direction.

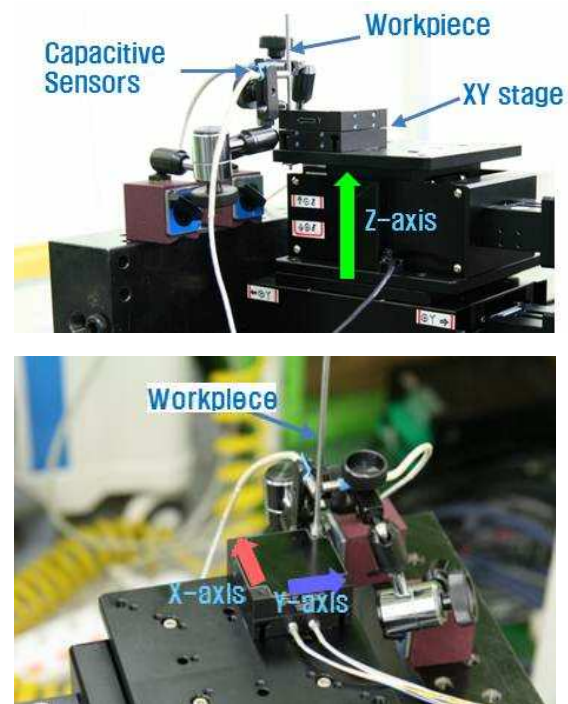
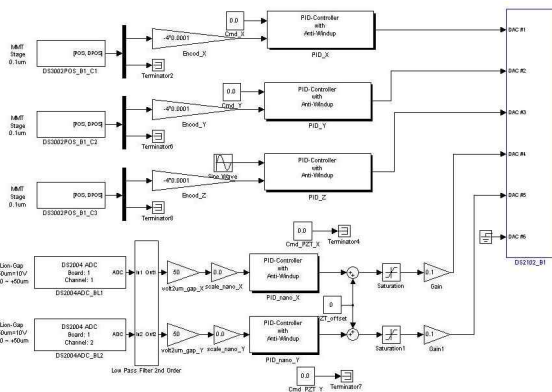
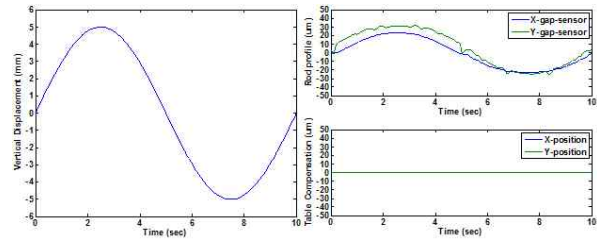


FIGURE 5. An experimental setup to compensate for positioning error of a cylindrical rod using two capacitive sensors in XY direction and an XY stage.

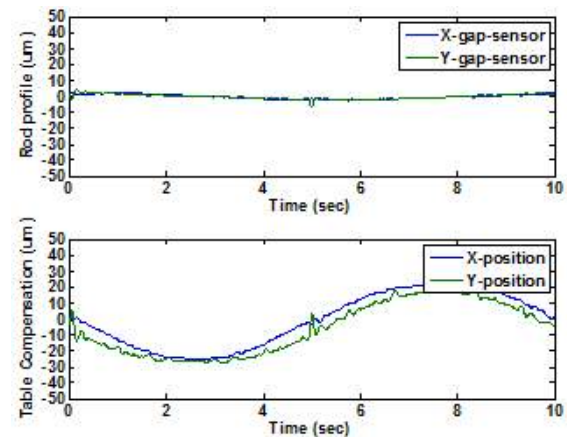
For real-time positioning error compensation, a dSpace control system was used with a 2 kHz sampling rate. The control block diagram is shown in Figure 6. There are five PID control loops in the control system. The first two PID controllers are connected to another stage to bring the XY stage within the working range of the capacitive sensors in the beginning and they have nothing to do with real-time compensation. The third one is for the Z-axis feeding. The fourth and fifth controllers drive XY stage to compensate for the positioning error based on capacitive sensor signals.



To verify the positioning error compensation performance, the Z-axis moves up and down over 10 mm distance with carrying a 3 mm diameter rod and an XY stage. As mentioned before, the capacitive sensors used in this experiment hardly measure the distance from a target smaller than 3 mm in diameter. Figure 7 shows the sensor signals when the compensation is turned off. The left plot shows the Z-axis motion and the upper right plot reveals the surface profile of the 3 mm diameter rod corresponding to the Z-axis motion. The XY stage is not moving as depicted in the lower right plot.



When the positioning error compensation is off, the sensor signal in X-axis shows a 46.0 μm Peak-to-Valley (16.5 μm RMS) variation, in Y-axis there is a 57.2 μm PV (21.2 μm RMS) variation along 10 mm distance in Z-direction. When the compensation is on, the sensor signals reduce to 4.3 μm PV (1.3 μm RMS) in X-direction and 11.7 μm PV (1.4 μm RMS) in Y-direction as shown in Figure 8. In terms of RMS values, they are 92.1% and 93.3% reductions, respectively.



Since the workpiece is a long and slender rod or a tube, it is easy to vibrate. It is important not to excite the resonant frequencies of the workpiece from the XY stage motion while high control

gains for the XY stage are required for agile responses from the stage.

CONCLUSION

To generate micro features on the surface of tube shape biomedical applications using laser machining, a positioning error compensation testbed was built and tested. Two sensors are adopted for experiments: edge detecting sensor and capacitive sensor. The edge detecting sensors may be used to measure the position of very small diameter workpieces, but it turned out to be vulnerable to contamination in real machining environments. The capacitive sensors are easy to install and safe against contamination, but have a limitation on target size. The testbed was built with two capacitive sensors in the XY plane. Using the gap sensor signals and high gain PID feedback loops, the positioning error reduced to 1.3 μm RMS and 1.4 μm RMS in X- and Y-axis, respectively, from 16.5 μm RMS and 21.2 μm RMS. The error reduction rate is more than 90% and the testbed setup is applicable to real laser machining.

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Effects of the tool edge shape on the brittle cracking in the glass machining

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1. INTRODUCTION

In recent years, the demand of the micro fabrication on brittle materials is increased. For example, a chemical inspection plate is widely used in chemical and pharmacy fields. If the accuracy is required, this device is produced with optical glasses because of their excellent mechanical or chemical properties. And the other example, a micro molding die is used to manufacture micro mechanical components. Tungsten carbides are used for this molding die to replace conventional hard steels because of its high stiffness and low thermal expansion. Generally, these devices are fabricated by the chemical etching or laser milling so far. However, these approaches have some problems. For example, in the chemical etching on glass materials, hydrofluoric acid which is required as the etching liquid has high toxicity. The laser milling induces the thermal deformation and changing the material property on the carbide surface. Also, both manners require the high cost for facilities and the manufacturing.

Based on this background, the author proposes the cutting process on these brittle materials with small diameter carbide ball end mills. The milling process on the brittle materials requires only the machine tool and cutting fluid (for example, machining center and distilled water) with safer and easier operations. In previous cutting experiments, brittle materials can be machined without brittle cracking in the proposed method. In these experiments, the PCD (Poly-Crystal-Diamond) coated carbide milling tool as shown in FIGURE. 1 achieves the high durability of the tool wear. In cutting tests, the tool life is 15 times longer than that of the TiAlN coated carbide tools. However, unfortunately, the many small cracks are left on the machined surface as shown in FIGURE. 2 at the same cutting experiments. This problem is regarded that the roughness of the tool edge shape changes the cutting thickness as shown in FIGURE. 3, and the stress distribution at the cutting point is also partially changed. Thus, the

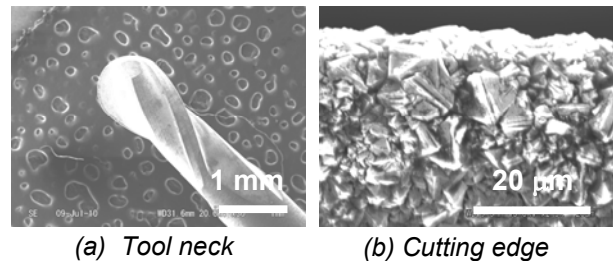


FIGURE 1. PCD-COATED CARBIDE BALL END MILL
Specifications: Material, carbide; coat, poly-crystal diamond; Radius, 0.5 mm; and helix angle, 25 deg.

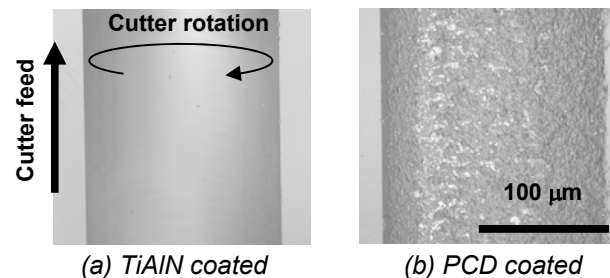


FIGURE 2. BRITTLE CRACKING ON THE SURFACE
Cutting conditions: Tool radius, 0.5 mm; tool helix angle, 30 deg; feed rate, 0.48 mm/min; rotational speed, 20000 rpm; and the depth of cut in Z axis of 0.02 mm.

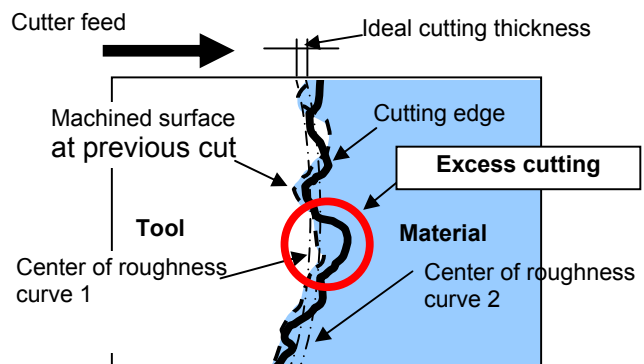


FIGURE 3. EXCESS CUTTING BY ROUGHNESS OF CUTTING EDGE

brittle cracking leave at the point of excess cutting thickness. According to this consideration, cutting experiments were performed with two kinds of PCD coated carbide milling tool they have different coating grain size. As the result,

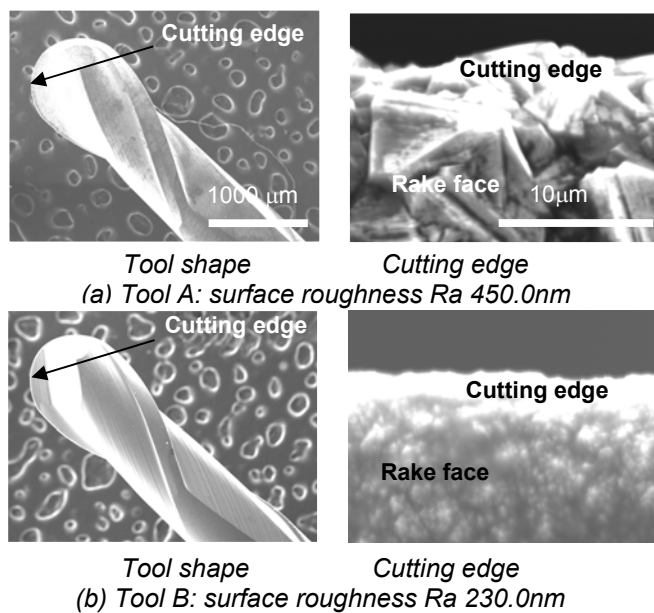


FIGURE 4. PCD COATED CARBIDE BALL ENDMILLS
 Specifications: Material, carbide; radius, 0.5 mm; neck length, 2.5 mm(a), 2.7 mm(b); and helix angle of 25 degree.

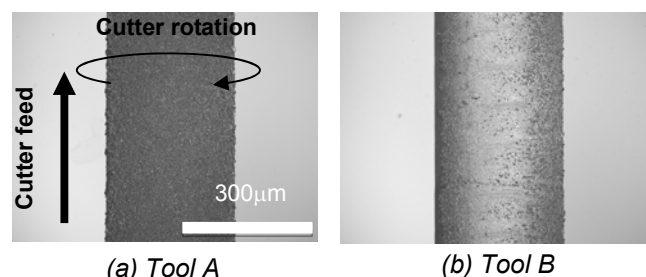


Fig. 5 BRITTLE FRACTURE IN MACHINED SURFACES
 Cutting conditions: Tool, R0.5 diamond coated ball endmill; material cut, soda lime glass; feed rate, 0.48 mm/min; rotational speed, 20000 rpm; feed rate per edge, 12 nm/edge; cutting speed, 62.8 m/min; depth of cut, 0.05 mm; and lubrication, water.

intensity of brittle cracking changes by the grain size as shown in FIGURE. 4^[1].

However, these results were not determined under quantitative conditions. Therefore, this phenomenon must be evaluated with quantitative cutting conditions. Therefore, the author has tried to observe effects of the roughness of the tool edge shape on the brittle cracking with shape controlled tools. In this research, the shape of the tool edge is designed as the simple sine wave to evaluate the characteristics of brittle cracking by features of edge shape such as amplitude, period or anymore. This report discusses about the effect

on the brittle cracking by the amplitude of the controlled edge shape in the glass cutting. Results of numerical simulations and cutting experiments by the shaping process with the shape controlled tools are introduced.

2. NUMERICAL SIMULATIONS

To evaluate the stress distribution of the material near by the cutting edge, following FEM simulations were performed by the ANSYS R11. In the actual cutting, the chip is removed with the dynamic road. However, the stress-strain characteristics of the optical glass in the cutting process have not been reported clearly because it is difficult to evaluate with experimental approaches such as hopkinson bar test. Therefore, in FEM simulations, it is assumed that the static deformation is applied on the material where the rake face of the cutting edge contacts.

FIGURE. 6 shows the specimen model in the simulation. The model of 20 mm in length and width, and 5 mm in height was determined as about 190,000 hexahedral solid elements. In the simulation, the model is assumed as the soda lime glass (73% SiO₂, 14% Na₂O, 9% CaO, 4% MgO, 0.03% K₂O). Material properties of the model are summarized in the Table 1^[2]. The model has the groove as the machined surface. FIGURE. 7 shows the schematic of the cross section of the groove. The groove has the sine wave surface. In simulations, the amplitude of the surface cross section A is varied of 0 (flat surface), 0.5, 1.0 μm to observe the effect on the brittle cracking. And the deformation of 10⁻¹⁰ m is applied on the side face of the cutting groove. The deformation is decided as the critical value where the principal stress rises over the fracture stress (≈ 35 MPa^[2]).

FIGURE. 8 shows the distribution of the principal stress and stress vector in simulations. In these FIGURES, the principal stress is described between -10 to 120 MPa. And yellow region describes the critical stress of the fracture (30-35 MPa), Orange region describes the may be fractured area (35-70 MPa), Red region shows the over the twice time of the fracture stress (> 70 MPa), respectively. As shown in FIGURE 7(a), in case of the flat surface (A = 0 μm), the stress distribution is uniform along the cutting edge and it is not above the critical stress. On the other hand, in the case of waved surface, the stress distribution concentrates on the bottom of wave, and it is increased with the amplitude A. Especially, in the case of large amplitude (A = 1 μm) the stress increases more

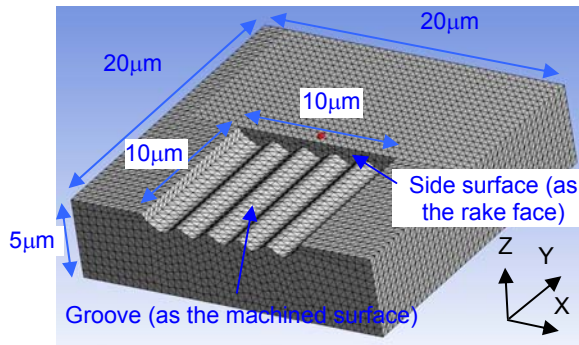


FIGURE 6. FEM MODEL

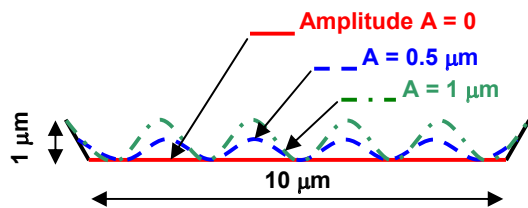


FIGURE 7. CROSS SECTION OF THE GROOVE IN FEM MODEL

TABLE. 1 MATERIAL PROPERTIES

Young module GPa	69.6
Poisson ratio	0.23
Density kg/m ³	2440.0
Yield Stress MPa	38.0

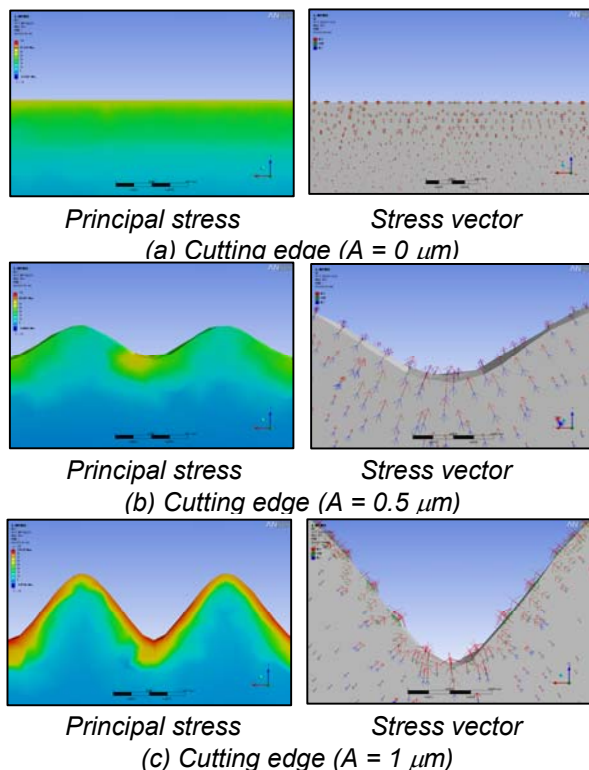
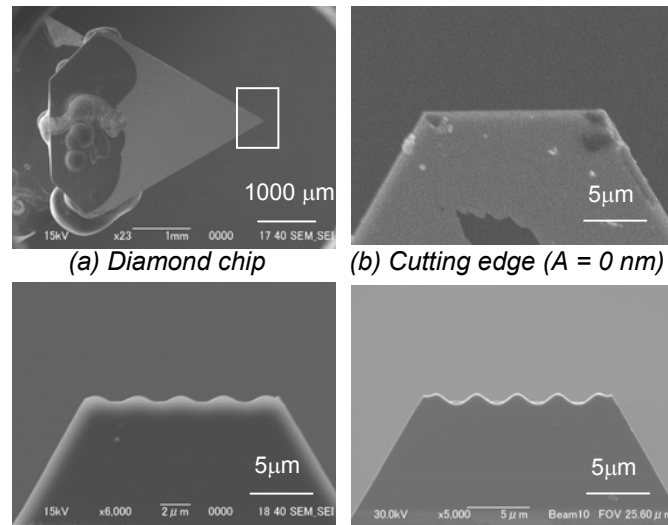


FIGURE 8. THE PRINCIPAL STRESS AND ITS VECTOR IN FEM SIMULATIONS



(c) Cutting edge ($A = 500 \text{ nm}$) (d) Cutting edge ($A = 1000 \text{ nm}$)

FIGURE 9. CUTTING EDGE SHAPE OF DIAMOND TOOLS
Specifications: Material, single crystal diamond; width of main cutting edge, 0.015 mm; rake angle, 0 deg; and the relief angle of 5 deg.

than 70 MPa that is twice time of the fracture stress.

3. CUTTING EXPERIMENTS

To evaluate the effect of tool edge shape on the brittle cracking, following cutting tests are performed. In cutting tests, slotting operations were performed with shaping process by the mono crystal diamond tool which is dressed as the same shape of the FEM simulations as shown in FIGURE. 9. FIGURE. 10 shows the machined surface in the cutting tests on soda lime glass with corresponding conditions of the simulation [3]. They are dressed by the Focused Ion Beam (FIB) machine tool. The shaping was performed by the liner motor stage (Nihon Thompson NT80V65). The resolution of the motor stage is guaranteed at 0.1 μm as the specification. This FIGURE shows the cross section of the machined surface at the depth of groove of 1 μm . Cross sections were machined by the FIB and taken by Scanning Micro Scope (SEM). The brittle cracking is growth corresponding to results of the FEM simulations.

The reason of these results is considered that the curvature of the sine waved notch is increasing with its amplitude, and this increasing makes the stress concentration at the top of the notch. These simulations are not same as the actual cutting phenomena completely, because they are only considered about the static deformation. However, their distributions are not

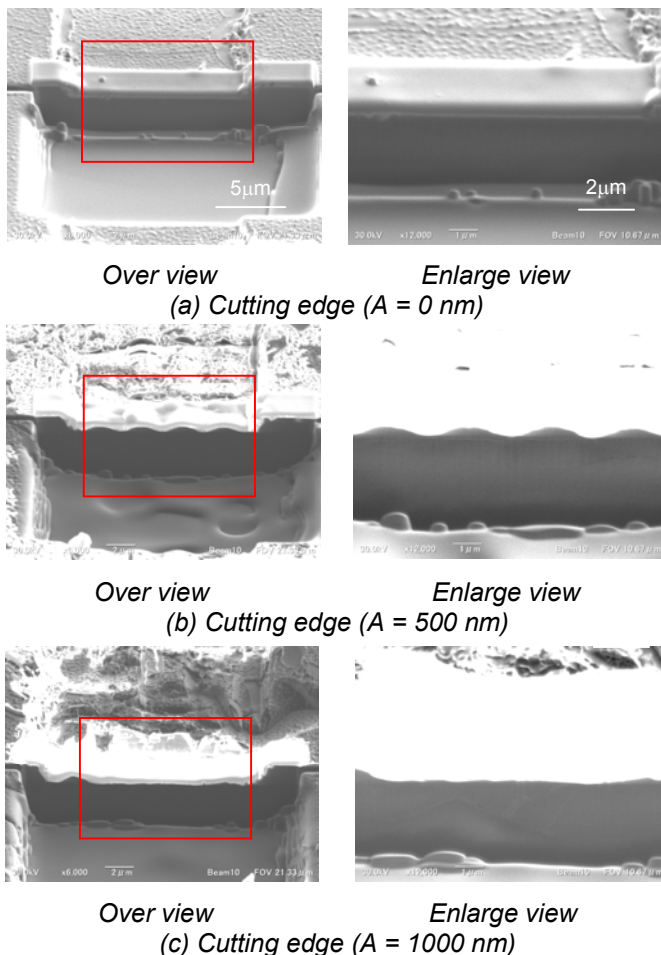


FIGURE. 10 CROSS SECTION UNDER THE MACHIEND SURFACE

Cutting conditions: Tool, Single crystal diamond tool; material cut, soda lime glass; Cutting speed, 1.0 mm/sec; and lubrication of dry.

far different from each other because the cutting tests were performed at the low cutting speed of 1 mm/sec. Therefore, this report concludes that the amplitude of the edge shape affects on the characteristics of the brittle cracking in the glass cutting because it changes the stress distribution of the material.

4. CONCLUSION

In this report, the effect of the cutting edge shape on the brittle cracking is discussed. The FEM simulations were performed to observe the stress distribution of the material near by the cutting edge. Results of simulations show the principal stress was increased with the amplitude of the edge shape that is designed as the cutting roughness. These results are

corresponding to cutting tests. The brittle cracking on the machined surface is grown with the amplitude of the cutting edge at same cutting conditions of simulations. Based on these results, this report conclude the amplitude of the edge shape affects on the characteristics of the brittle cracking in the glass cutting because it changes the stress distribution of the material.

ACKNOWLEDGMENT

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FEM ANALYSIS OF THE EFFECT OF SEQUENTIAL CUTS ON HEAT PARTITION IN HIGH-SPEED CUTTING

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INTRODUCTION

In metal cutting, mechanical energy is transformed into heat through the plastic deformation in chip formation and the frictional work between the tool and the workpiece. As a result, a portion of the generated heat is contained by the chips, while the heat flows into the workpiece and the tool will cause the temperature rise. High cutting temperature causes a short tool life, as well as poor surface finishing. Thus, it is important to quantitatively elucidate the correlation between these thermal behaviors and machining errors. A finite element analysis is available to simulate the cutting process and can predict the temperature distribution, cutting force, tool wear and residual stresses, etc. Furthermore, by taking advantage of FEM analysis of temperature distribution in the workpiece, tool and chip, the tool geometry and cutting conditions, which will minimize the temperature rise and achieve a high cutting accuracy, can be optimized. However, to analyze the temperature distribution during cutting, the heat partitions flowing into the workpiece, tool and chip have to be determined. In this paper, a two-dimensional FEM analysis model was developed to determine these thermal partitions.

On the other hand, in some cutting processes, the affected layer and temperature rise area of the workpiece may be subjected to the following cutting. For example, in the milling, the machined layer by the preceding cut will be cut again by a following cutting edge over and over. In this case, the surface performance of the workpiece is clearly affected by the final cutting. This cutting situation is known as "Sequential Cutting". Several researchers have devoted their efforts to investigating the residual stress distribution in sequential cuts[1]-[5]. Sasahara et. al.[1] investigated the effect of cutting sequence on residual stresses in machining of brass. Liu et.al.[2] developed a thermal-elastic-plastic model to investigate the effect of sequential cuts and tool-chip friction on residual stresses. On the other hand, there are some reports studying on

the cutting temperature[6]-[8]. While, the effect of the "sequential cuts" on the temperature distribution and thermal expansion of the workpiece has received little attention. In this paper, by taking advantage of the resolved heat partitions flowing to the workpiece, tool and chip under various cutting conditions, the temperature distributions and thermal expansion of the workpiece during the sequential cutting process were also investigated by FEM analysis. Especially, the effects of the time interval between two cuts on the temperature distribution in the workpiece were investigated.

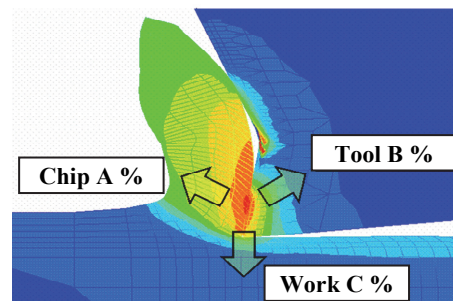


FIGURE 1. Heat partition flowing into workpiece, tool and chip.

CALCULATION OF HEAT PARTITION

In the cutting process, a portion of the generated total amount of the heat is always contained by the chip. The remainder of the heat flows into the workpiece and the tool, as shown in Fig.1. It is well known that the heat which flows into workpiece can be calculated as the temperature change of the workpiece, the specific heat of the workpiece material and the mass of the workpiece based on the following relationship:

$$\Delta Q = mc\Delta T \quad (1)$$

where ΔQ is the heat flows into the workpiece (J), ΔT is the temperature change in the workpiece ($^{\circ}\text{C}$), m is the mass of the workpiece (kg), and c is the specific heat of the workpiece ($\text{J/kg}\cdot\text{K}$). Similarly, if the temperature distributions in the tool and the chip are captured, the heat flows into the tool and chip also can be calculated from the above relationship.

ANALYTICAL PREDICTION OF SEQUENTIAL CUTTING PROCESS

Simulation Model

A turning and a milling process are commonly used for metal cutting. As a characteristic of the cutting operations, the layer of the workpiece affected by the first cutting is usually subjected to further cutting. Residual stress and strain, for example, will be affected by a cutting sequence. Similarly, the temperature distribution in the machined surface layer of the workpiece is also clearly affected during the sequential cuttings. However, from the point of view of the heat flow, it may be anticipated that, if the time interval between the two cutting paths is short enough, the heat generated from the first cutting can be altogether removed with new chips by the following cutting before the heat flows into the workpiece. Then it is expected that the temperature rise and thermal deformation of the workpiece will be also reduced.

In order to investigate the relationship between the time interval in the cutting sequence and the temperature distribution in the machined surface layer of the workpiece, a simulation of the sequential cutting process was carried out by taking advantage of a resolved heat partition.

Effect of time interval of cutting sequence as regards the temperature distribution and heat partition

Of the cutting parameters in the sequential cutting, the time difference between the first and second cuts are considered to play an important role in addition to the other cutting conditions, such as the cutting speed, the depth of cut, the tool material and tool shape. Three kinds of time intervals were applied in the analysis. Cutting conditions are shown in table 1. The time intervals are assumed from 0.2ms to 24ms. 0.2ms and 3.0ms are assumed based on the high-speed milling with various tool diameter and number of flutes, and 24ms is assumed based on the turning process. Both are realistic setting values.

The depths of cut of both the 1st and 2nd cuts in the sequential cutting were 0.03mm. Furthermore, in order to clearly the difference in temperature distribution in the workpiece between the single cutting process and the sequential cutting process, a temperature analysis of a one-path cutting process was also carried out for a comparison. The depth of cut on the one-path cutting was 0.06mm.

TABLE 1. Correspondence of cutting model and cutting conditions in FEM simulation.

Cutting model		High speed milling		Turning
		0.2	3.0	
Time interval	ms	0.2	3.0	24.0
Cutting speed	m/min	900		
Depth of cut	mm/tooth	0.03		
Tool / Workpiece diameter	mm	2	60	115
Number of flutes		2	4	1
Rotational speed	min ⁻¹	150000	5000	2500

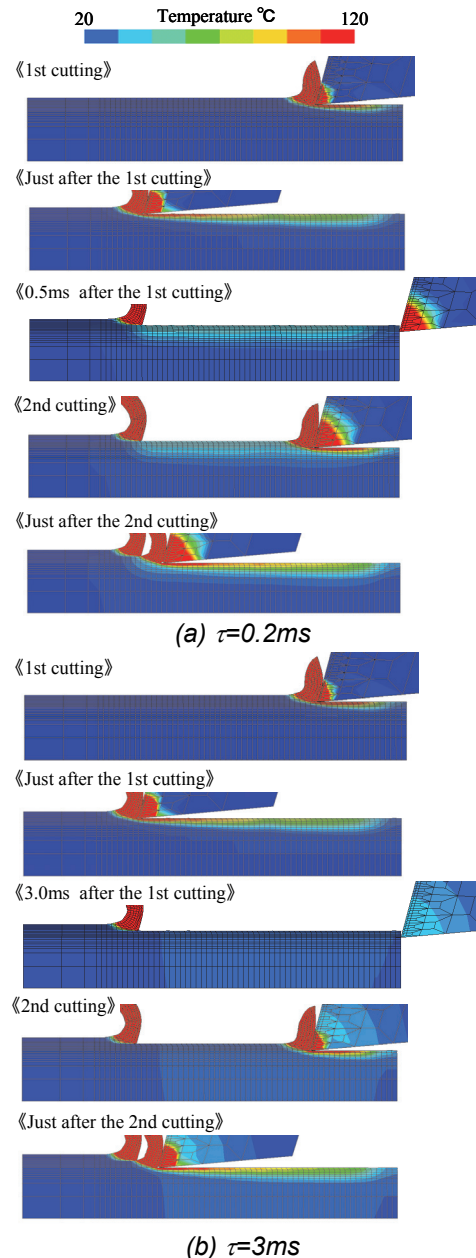


FIGURE 2. Simulated results.

Figure 2 shows the simulated results. For the cutting sequence, two cutting processes were carried out. First, a temperature distribution in the machined layer of the workpiece is formed through the plastic deformation and friction at tool-chip interface during the first cut. Then, the temperature field changes with the heat conduction inside the workpiece. Then, the time interval between the first cut and the second cut goes by. It can be seen that the time interval becomes longer, the temperature of the finished surface layer becomes wider because of the heat conduction. Finally, after the second cut, the top surface generated by the first cut is removed and then a new temperature field in the workpiece was formed.

Figure 3 shows the change of temperature within the machined surface, (a), (b) and (c) are the results of the sequential cutting with 0.2ms, 3.0ms and 24.0ms time interval respectively, (d) is the result of the just one-path cutting with depth of cut 0.06mm for the comparison.

In the sequential cutting processes, after the 1st cutting, the top surface layer is removed, and the temperature rises only in the narrow machined surface region. Then, in a short time, the heat conducts into the workpiece, the top surface temperature decreases and the temperature inside of the surface slightly increases as shown in green line (solid square symbol). Then the area of temperature rise will be quickly removed by the following cutting in the sequential cutting process. Finally the 0.03mm of the workpiece surface is removed, the temperature of machined surface rises as shown in red line (solid triangle symbol). It can be seen that the temperature rise region is limited within 0.15mm from the new surface

When the time interval is longer, the temperature distribution of the workpiece will be different. Fig.3 (b) shows the result with time interval τ is 3.0ms. Initial state and the 1st cutting process is just the same as shown in Fig.5 (a). However, after the longer initial time of 3.0ms, it is long enough to diffuse into the whole inside of the workpiece. Then the temperature within 0.15mm from the surface shows about 28 °C. It means that temperature rise of the whole workpiece is about 28 °C. Finally, the 2nd cutting is conducted, the workpiece temperature rises as shown in red line (solid triangle). However the temperature of the inside of the workpiece is higher than the case of the time interval is 0.2ms.

Then the difference between sequential cutting and one-path cutting will be discussed next. After the machining, the temperature rise in the workpiece in the one-path cutting (D : 0.06mm) is bigger than those in the sequential cutting (D : 0.03mm). Thus, in particular τ is 0.2ms, it can be anticipated that the thermal deformation of the entire workpiece caused by the temperature rise in the sequential cutting process is smaller than that in the one-path cutting. In simple terms, an

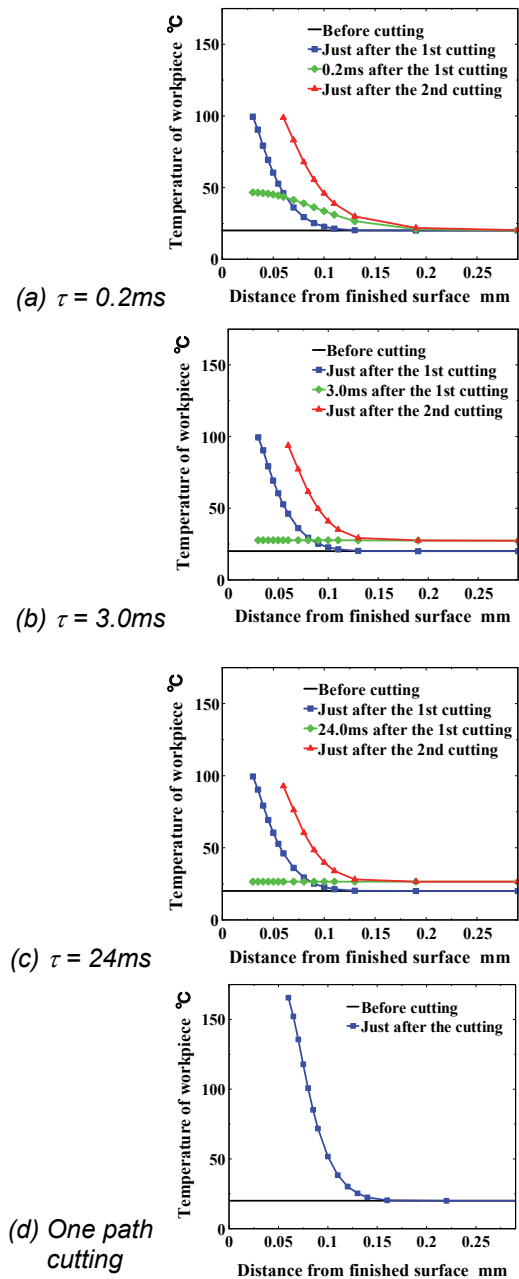


FIGURE 3. Temperature distributions inside the workpiece.

increase in the time interval will weaken the advantage of sequential cutting.

Figure 4 shows the amount of thermal energy flowing into tool, chip and the workpiece. It is apparent from the results that the total heat generated during the double sequential cutting with depths of cut of 0.03mm is larger than that in the one-path cutting with depth of cut of 0.06mm. However, according to the above considerations, when τ is 0.2ms or 0.5ms, the heat generated by the first cutting can be simultaneously removed as the new chips before the heat flows into the workpiece. Thus, actually, the heat flowing into the workpiece is smaller than that in the one-path cutting.

Figure 5 shows the thermal energy flowing into the workpiece in various time intervals. Then, it is understood that the heat that flows into the workpiece decreases rapidly when time interval goes shorter than 3ms. In other words, a lot of heat on the finished workpiece surface layer which is generated during preceding cutting is removed with the chip during the following cutting, in this order of time interval.

CONCLUSION

The cutting energy flowing into the chip, workpiece and tool were studied by a two-dimensional FEM analysis in this paper. In particular, the influences of effect of the sequential cutting were examined. When the time interval τ is longer than 3ms the energy flowing into the workpiece is almost constant. While, as the time interval τ becomes shorter than 3ms, the energy flowing into the workpiece decreases remarkably. It is because the high temperature layer on the surface of the workpiece generated by the preceding cutting can be removed immediately by the following cutting.

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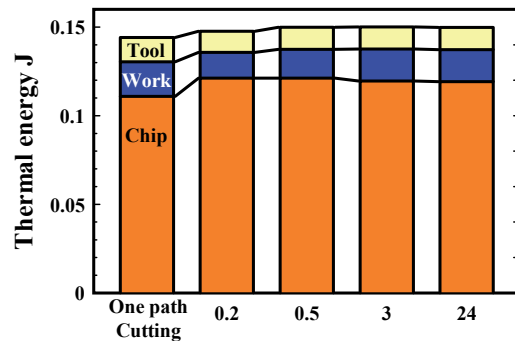


FIGURE 4. Thermal energy in various time intervals of sequential cutting process.

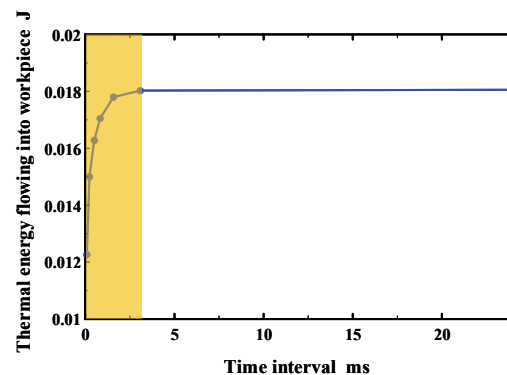


FIGURE 5. Effect of time interval on the thermal energy flowing into the workpiece.

- 28: 179-184.
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PRECISION GRINDING OF LARGE ASPHERIC GLASS LENSES FOR DIGITAL SINGLE LENS REFLEX CAMERAS

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INSTRUCTION

Needs of digital single lens reflex (SLR) cameras are increasing rapidly in recent years, and the demands of large aspheric glass lenses are increasing. The small glass lenses for the compact digital cameras are molded effectively by the glass press molding process with micro ceramics molds as shown in Figure 1(a), On the other hand, in the case of manufacturing large aspheric lenses made of glass such as the digital SLR camera, a series of grinding and polishing process, as shown in Figure 1(b) is more effective and more precise than the molding process. Precision grinding and polishing technologies for the complex shape of the aspheric lenses are required furthermore.

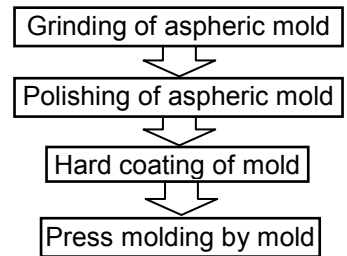
In this study, a new effective and precision grinding method is proposed and developed for the aspheric convex lens. The disk type of resinoid bonded diamond wheel which edge is trued to flat shape on the machine by a metal alloy, is scanned and controlled in 3-axis (X, Y, Z) and a contact point on the grinding wheel to the workpiece is changed in order to reduce the tool local wear. Contemplated by the proposed method can reduce the non-machining time due to wheel wear, and it is possible to improve the grinding efficiency.

CONVENTIONAL 2-AXIS CONTROLLED ASPHERIC GRINDING METHOD

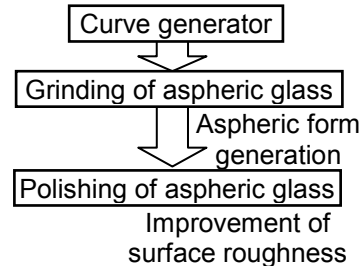
At first the glass material is pre-ground to an approximate radius curvature by the curve generator. In the conventional finish aspheric grinding process, the designed aspheric shape is generated by the CNC aspheric grinding machine with a disk type of the diamond wheel having a round edge as shown Figure 2.

The disk type wheel scans in 2 axis (X, Z), and the wheel will wear at the top of the wheel after several decades of grinding pass. Then the workpiece form error will occur in the center area of workpiece by the local wear of the wheel as shown in Figure 3. Figure 4 shows a

Nomarski micrograph of the error image in this typical ground workpiece center.



(a) Press molding process



(b) Direct machining process

FIGURE 1. Mass-production process of aspheric glass lens

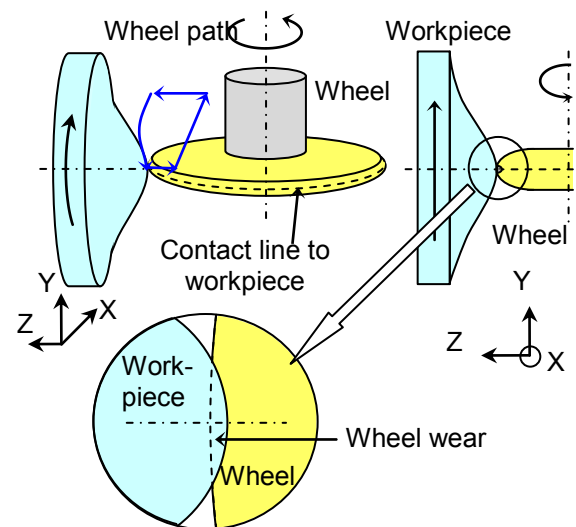


FIGURE 2. Conventional aspheric grinding method of 2 (X,Z) axis

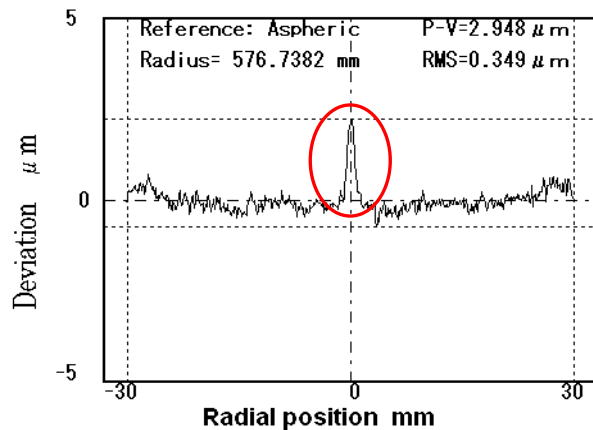


FIGURE 3. Typical form error by the local wheel wear in the conventional aspheric grinding



FIGURE 4. Nomarski micrograph in the center deviation image of the typical ground workpiece

PROPOSED ASPHERIC GRINDING METHOD

In this paper, a new aspheric grinding method of 3-axis control is proposed, as shown in Figure 5. A flat disk shape of the wheel is used, the wheel scans in 2-axis (X, Z), and after the pass, the wheel moves in Y-axis and the grinding point on the wheel changes every grinding path in order to reduce the wheel wear.

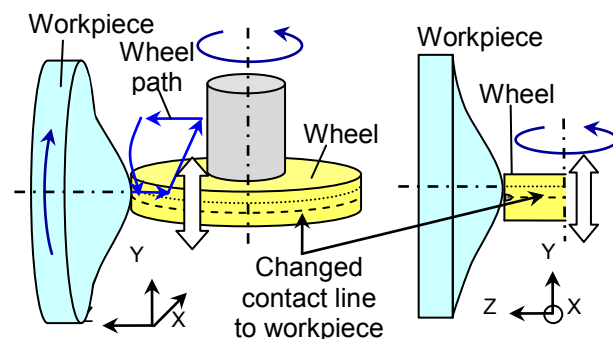


FIGURE 5. 3-axis (X,Y,Z) controlled aspheric grinding method

HIGH PRECISION AND HIGH PERFORMANCE TRUING / DRESSING OF DIAMOND WHEEL BY ALLOY METAL

In the aspheric grinding, the grinding wheels must be formed on the machine in order to reduce the run-out of the wheel and improve the workpiece form accuracy. After truing, the formed wheel surface must be dressed on the dressing process in order to sharpen the grain edges of the abrasives and increase the acting abrasive number on the wheel surface. In the conventional truing/dressing process of the resinoid bonded diamond wheel, the wheel is trued by a single crystalline diamond truer and the trued diamond wheel is dressed by a green silicon carbide (GC) stone, as shown in Figure 6. However, in the conventional truing process, the diamond abrasives on the surface will be dropped off by the diamond truer and the acting abrasive number on the wheel surface will be decreased. In the conventional dressing process of the resinoid bonded diamond wheel, the abrasive on the wheel surface will be sharpened and the number of acting abrasive grains will increase on the wheel surface by rubbing by the green silicon carbide (GC) stone. However, it is difficult to sharpen the wheel edge, because the GC stone is not so hard compared with the diamond wheel.

This paper proposed that the alloy metals are used as a truer/dresser of high precision and high performance. Alloy metals are expected to be excellent tools for diamond, because they seem to be active to the carbon atom of diamond. In order to improve the grinding accuracy, the wheel side surface is rubbed with the alloy metal and is formed precisely on the machine, as shown in Figure 7.

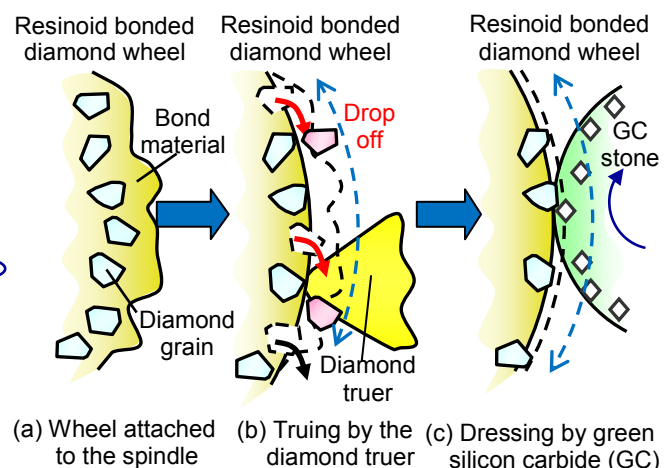


FIGURE 6. Conventional truing / dressing process for resinoid bonded diamond wheel

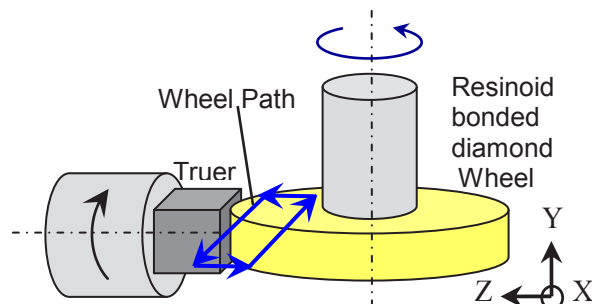


FIGURE 7. Proposed truing method of flat wheel

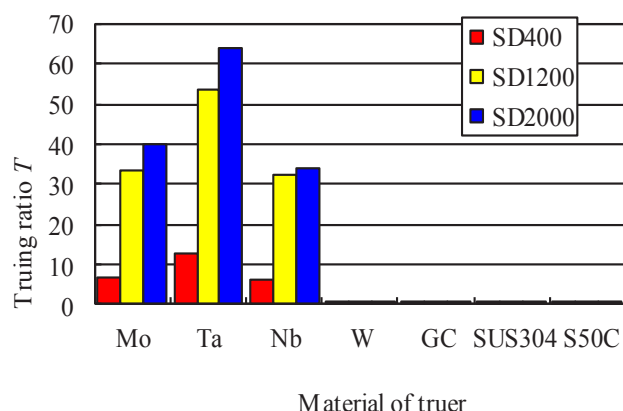


FIGURE 8. Change of truing ratio in each truer

Truing efficiency, i.e. the ratio of trued wheel volume / worn truer volume, is important in improving a flatness of the grinding wheel and reducing the total truing time. In this experiment, the truing ratio is evaluated by using some expected truer materials. A truing ratio is defined by the equation: $T = V_e / V_t$ (1) Where, V_e is a trued wheel volume and V_t is a worn truer volume. This value, T becomes larger when the truing efficiency becomes higher. Figure 8 shows changes of truing ratio in each truer. In case of using Mo, Ta and Nb, the truing ratio, or the truing efficiency was so much higher than the conventional materials. Cylindrical diamond wheel was trued by each truer material. Depth of cut was 1 μm and the truing pass number was 500. This seems to be caused that those alloy metals are active to the carbon atom of diamond.

ASPHERICAL GRINDING METHOD

Figure 9 shows the ultra-precision machine used in the experiment. The grinding machine was 4-axis (X,Y,Z,C) controlled and workpiece spindle and grinding spindle are polous type of air bearings and the guides were roller type slide

tables. And Figure 10 shows the grinding set-up. Grinding conditions are shown in Table 1. The workpiece was an optical glass and the wheel was resinoid bonded diamond of #1200 in grain size. Every depth of cut was 3 μm . After every 10 grinding pass, the ground workpiece surface was measured by contact type of measuring instrument "Form Talysurf" to evaluate the changes of the form deviation and the tool wear.



FIGURE 9. View of 4 axis (X,Y,Z,C) controlled ultra precision grinding machine

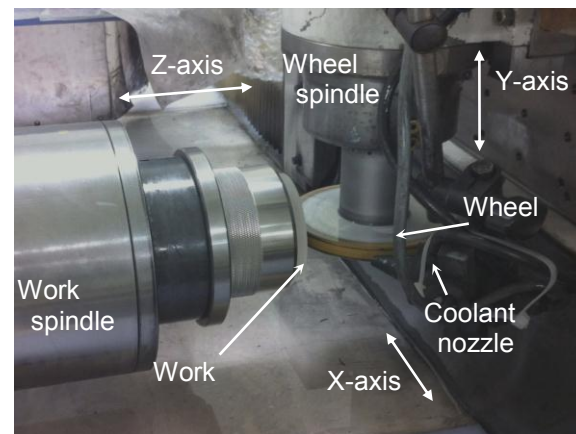


FIGURE 10. A view of aspheric grinding

TABLE 1. Grinding conditions

Workpiece	Glass
Diameter	$\Phi 60$ mm
Approximate radius	550 mm, Convex
Rotational speed	500 min^{-1}
Wheel	Resinoid bonded diamond
Grain size	1200
Diameter	$\Phi 100 \text{ mm} \times 5 \text{ mm t}$
Rotational speed	$15,000 \text{ min}^{-1}$
Depth of cut	3 μm
Feed rate	5 mm/min
Coolant	Solution type

ASPHERICAL GRINDING RESULTS

The tool radius error, ΔR and the tool positioning error, ΔX were evaluated from the measured initial grinding results to compensate the form accuracy. Figure 11 shows a form deviation profile which was ground by the proposed method and compensated the tool errors. The form deviation was improved to $0.81 \mu\text{m}$ P-V. Figure 12 shows a change of form deviation in the ground workpiece. In the proposed grinding method, the wheel height was changed in Y-axis direction. Very fine form deviations have been kept after 70 pass of grinding, because the local wheel wear was decreased so much by the proposed and developed method.

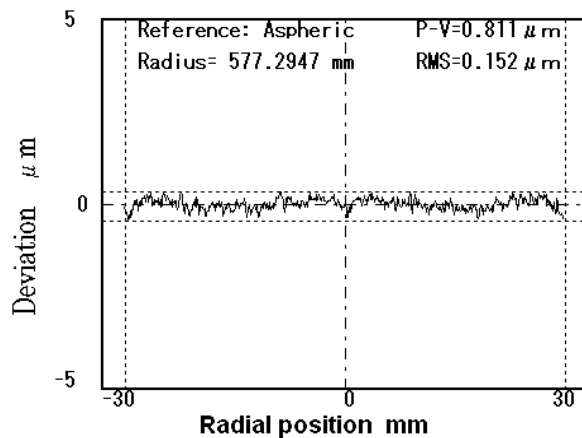


FIGURE 11. Form deviation profile by the proposed grinding

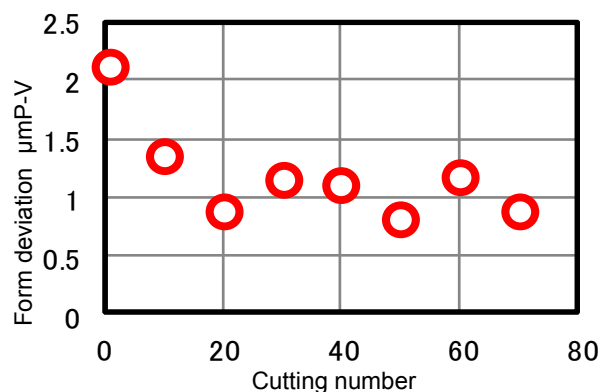


Figure 12: Change of form deviation

CONCLUSIONS

In this study, a new effective and precision grinding method is proposed and developed in order to grind large aspheric glass lenses

precisely and effectively. The disk type of resinoid bonded diamond wheel which edge is trued to flat shape on the machine by a metal alloy, is scanned and controlled in 3-axis (X, Y, Z) and a contact point on the grinding wheel to the workpiece is changed in order to reduce the tool local wear. From the aspheric grinding experiments, it was cleared that the proposed grinding methods are useful for grinding the large aspheric glass lenses directly.

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UNIFORM POLISHING OF LARGE ASPHERIC GLASS LENSES BY MAGNETIC FIELD-ASSISTED POLISHING

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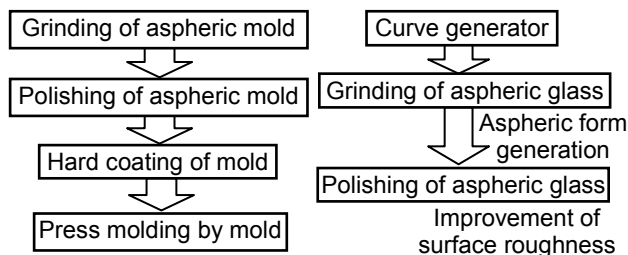
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INTRODUCTIONS

Recently, demand is increasing for digital high-quality single-lens reflex (SLR) camera, which can take sharp and clear photographic images compared to a compact camera. As a pixel number of the CCD image sensor is more and more increasing, the preciseness of the large aspheric glass lenses is more and more required and lens size is also required to be increased. In order to remove the chromatic aberration, it is effective to counteract the change of refractive index by combining the convex lens and concave lens.

The small glass lenses for the compact digital cameras are molded effectively by the glass press molding process with micro ceramics molds as shown in Figure 1(a) [1-3]. On the other hand, in the case of manufacturing large aspheric lenses made of glass such as the digital SLR camera, a series of grinding and polishing process, as shown in Figure 1(b) is more effective and more precise than the molding process. Precision grinding and polishing technologies for the complex shape of the aspheric lenses are required furthermore.

As a finish process of the ground glass lenses, a uniform polishing by using a balloon type of polisher with high pressed air is useful in case of convex lenses of flat workpieces. However, in case of polishing concave lenses, it is difficult to polish uniformly [4-5].

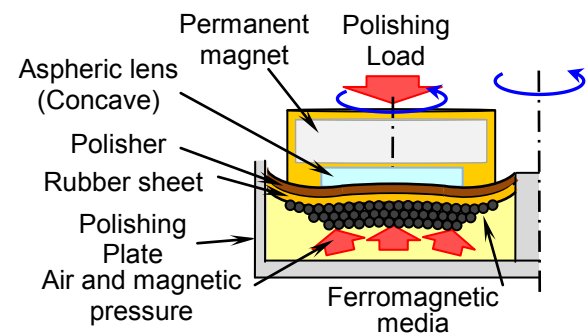


(a)Molding process (b)Direct machining process
FIGURE 1. Mass-production process of aspheric glass lens

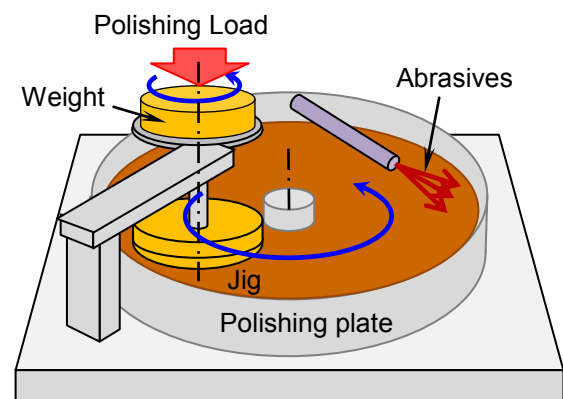
In this study, a magnetic field-assisted polishing using a permanent magnet and ferromagnetic media was proposed and developed in order to polish the large concave aspheric glass lenses smoothly and efficiently.

POLISHING PRINCIPLE

Figure 2 shows a schematic illustration of the developed magnetic field assisted polishing. A groove cut on a rotational stainless steel disk is sealed with a rubber sheet and polisher, and this groove is filled with ferromagnetic media. The permanent magnet is installed in the polishing jig and the workpiece is attached onto the jig. An aspheric lens contact to the polisher and the



(a) Mechanism of uniform polishing pressure



(b) Polishing method of concave lens

FIGURE 2. Proposed uniform polishing of magnetic-field assisted polishing

polisher presses the workpiece surface by the magnetic pressure caused by the magnet and ferromagnetic media. The polishing plate rotates and the workpiece also rotates in the same direction. This sun-and-planet motion makes the workpiece surface polished with polishing slurry. If the distribution of the polishing pressure is uniform, the removal distribution is also equal.

POLISHING MACHINE

Figure 3 shows the magnetic assisted polishing machine developed for aspheric uniform polishing of ground glass lenses. The lens is attached onto the polishing jig and a center of the jig is supported with a weight bar and polishing load is applied to the jig with a magnet. The ferromagnetic media under the polisher also applies the polishing load generated by the magnetic field. The workpiece will rotate at the same speed and the same direction as the polisher, and the relative speed is the same in the workpiece surface. Then keeping the polishing pressure distribution equal is most important to do equal polishing.

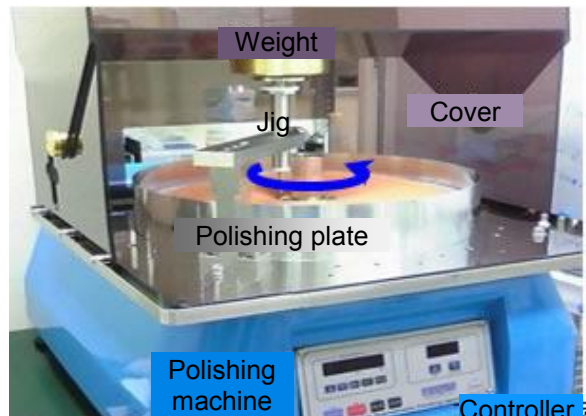


FIGURE 3. Experimental polishing set-up of the magnetic field assisted polishing

EXPERIMENTAL METHOD

In the experiment, flat BK 7 glass was polished in order to evaluate the basic polishing characteristics. Polishing conditions are shown in Table 1. As a polisher, urethane pad of 1.0 mm thickness was used. As an abrasive, slurry of cerium oxide of 10 %wt was used. The peripheral speed was 9.99 m/min. Changes of polishing removal and surface roughness were evaluated. Form accuracy was measured by non-contact type of blue laser probe NH-3UP, Mitaka kohki Co.Ltd [5]. Surface roughness was measured by non-contact three-dimensional

scanning white interferometer New View 6200, ZYGO.

TABLE 1. Polishing conditions

Lens material	BK7
Diameter	Φ80 mm
Approximate radius	R280 mm
Polymer	Urethane pad
Thickness	1.0 mm t
Abrasive	Cerium oxide
Density	10 wt%
Polishing pressure	4.97 kPa (50.7 gf/cm ²)
Rotation speed	15 rpm
Radius of polishing point	r=106 mm
Polishing speed	9.99 m/min

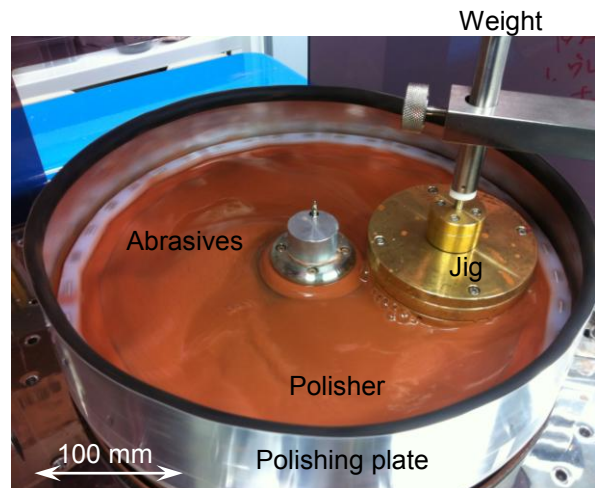
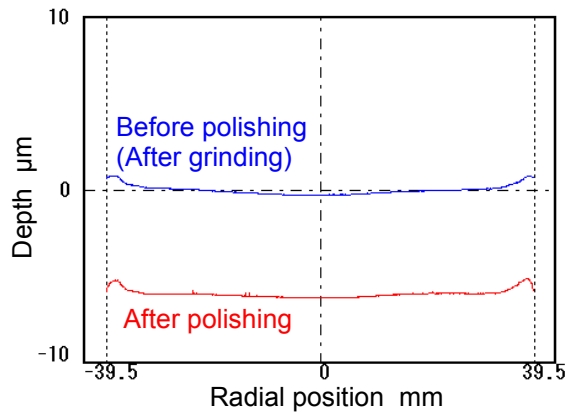


FIGURE 4. View of uniform polishing

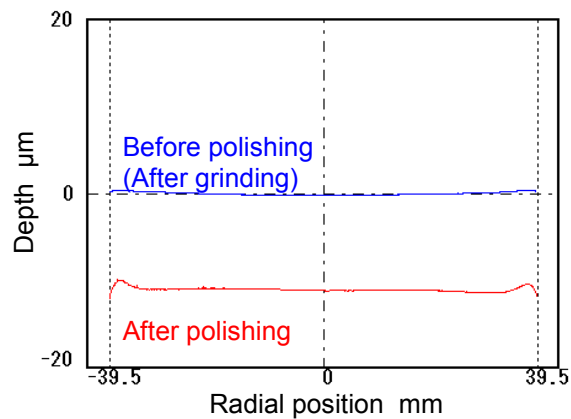
EXPERIMENTAL RESULTS

Figure 5 shows changes of form deviation profiles before and after polishing, measured by the non-contact blue laser probe. (a) is after 25 minutes and (b) is after 60 minutes. Figure 6 shows polishing removal distributions, calculated from the digital data of the measured profiles in Figure 5. Depth removal by 60 polishing was 11 μm. Polishing uniformity was 89 % and uniform polishing was carried out.

Figure 7 shows a Nomarski micrograph and a surface roughness profile after 25 min polishing. Figure 8 shows a Nomarski micrograph and a surface roughness profile after 25 min polishing. Figure 9 shows changes of depth removal and surface roughness. The surface roughnesses were improved with the polishing time. Very smooth surface were obtained.

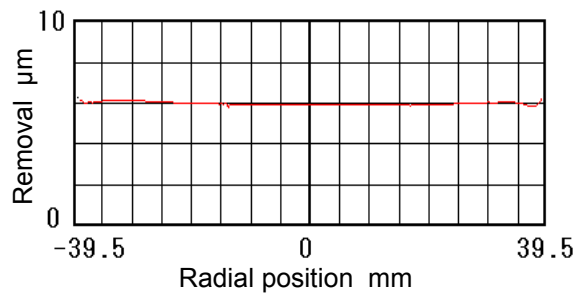


(a) After polishing of 25 minutes

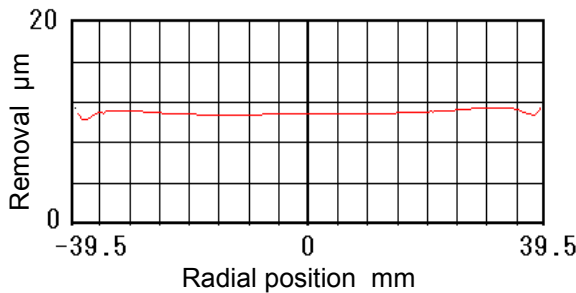


(b) After polishing of 60 minutes

FIGURE 5. Changes of deviation profiles



(a) After polishing of 25 minutes

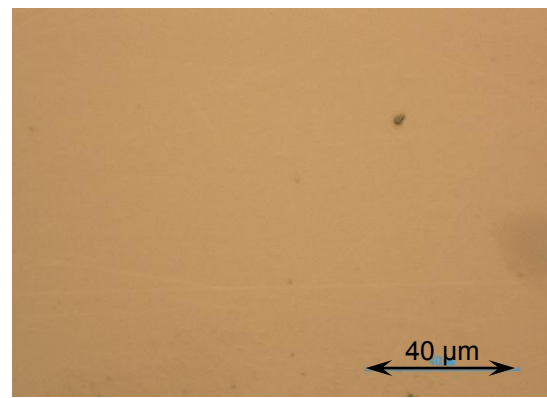


(b) After polishing of 60 minutes

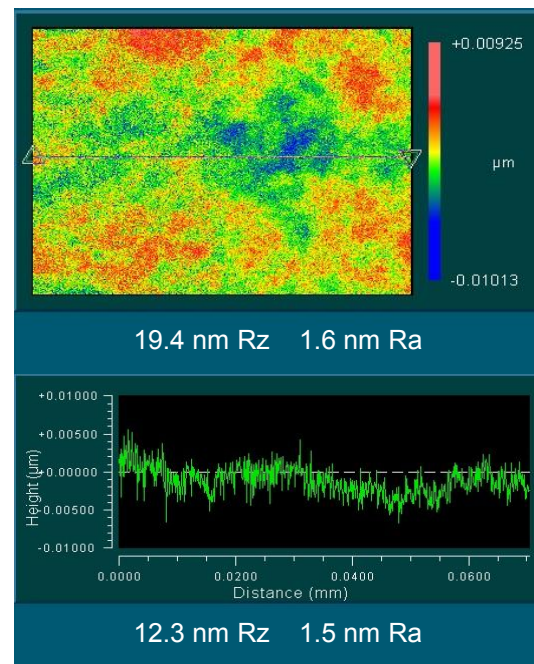
FIGURE 6. Removal distributions

CONCLUSIONS

In this study, the magnetic field-assisted polishing was proposed for high effective and high precision polishing of the large aspheric glass lenses, and uniform polishing machine was developed by using a permanent magnet. In the polishing experiment of the BK7 glass, changes of the removal distribution and surface roughness change were evaluated. From the experimental results, it was clarified that the proposed uniform polishing methods are useful for the finishing of large concave aspheric glass lenses.

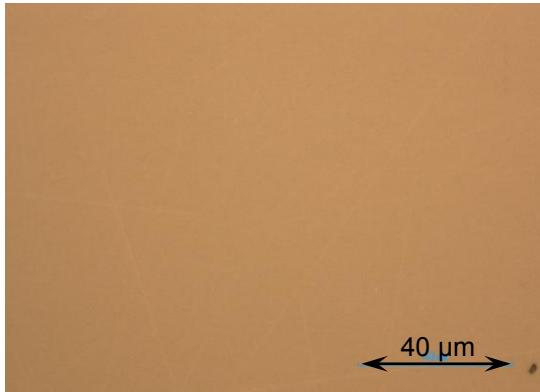


(a) Nomarski micrograph of polished surface

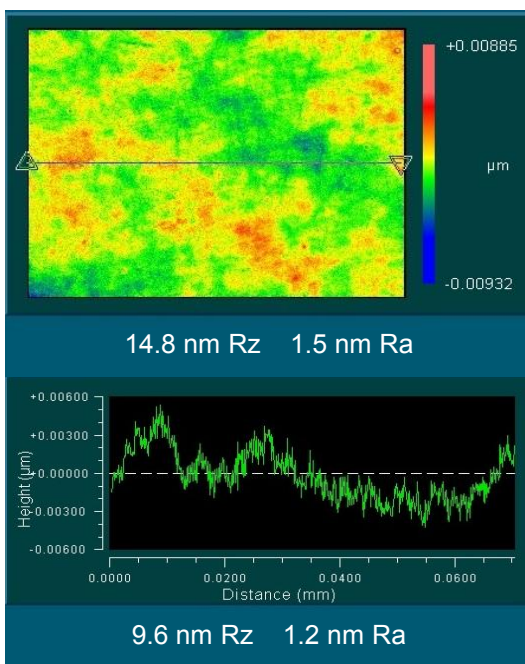


(b) Surface roughness profiles

FIGURE 7. After polishing of 25 minutes



(a) Nomarski micrograph of polished surface



(b) Surface roughness profiles

FIGURE 8. After polishing of 60 minutes

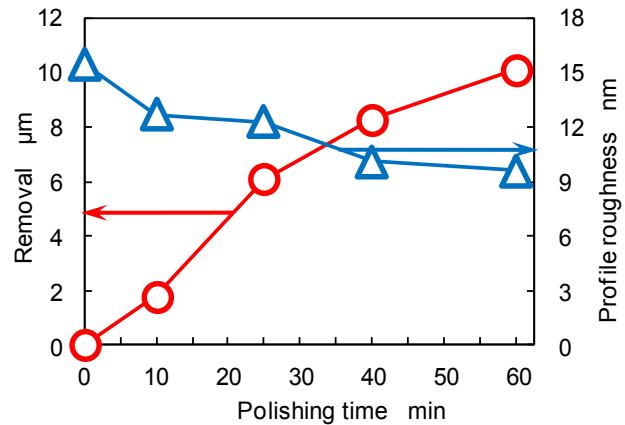


FIGURE 9. Changes of depth removal and surface roughness

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SIMULATION STUDIES OF SUB-MICROMETER CONTACT OF TOPOGRAPHY STYLI

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BACKGROUND

The contact of small probe-tip against a surface is crucial to stylus-based surface roughness metrology, which remains a popular and important practical technique for both industry use and basic research. This discussion paper focusses on simulation studies intended to improve metrological understanding (although it might have other micro-tribology applications). A finite stylus reduces measurement fidelity through direct geometrical effects, elastic distortion and dynamical effects that vary the contact force, sometimes to lift-off.

Obviously, surface features with dimensions, curvatures, etc. smaller than or even somewhat larger than those of the tip will be distorted during measurement. Therefore, simulations concentrate on how the stylus contact modifies the truly existing topography. Except for some statistics reported in [1], they tend to ignore the effects on the stylus tip itself. Appropriate 3D simulations have awaited adequate low-cost computing power. It is now practicable to run numerical experiments that may: cast light onto questions of in-plane uncertainty of stylus measurement; suggest improved ways to compare between different instrument probing technologies; or lead to methods for self-diagnosis of stylus wear [2].

THE NEW STEPS

This presentation reports recent progress on extended studies into the contact regimes of surface profilometer styli, with a view especially to better understanding the uncertainty budgets [3]. It simulates the 3D contact of an approaching (ideal or real) stylus. Behaviour at the limits of instrument capability is studied by using AFM-derived data to represent small areas of engineering surfaces at resolutions to 1 nm vertically and 200 nm laterally. Extending upon conventional kinematic simulations, its novelties include (i) detailed analysis of how contact points or patches scatter with respect to

the normal axis and (ii) a “threshold” function for exploring numerical and physical sensitivity to very small surface features (potentially noise).

Discussion will summarize the main features of the software implementation (using commercial packages MATLAB and SPIP), the rationale for statistical parameters to summarize contact spatial distribution, and the use of these parameters to demonstrate the in-plane uncertainty to be expected when scanning fine ground surfaces with a typical (2 μm) stylus.

RESULTS

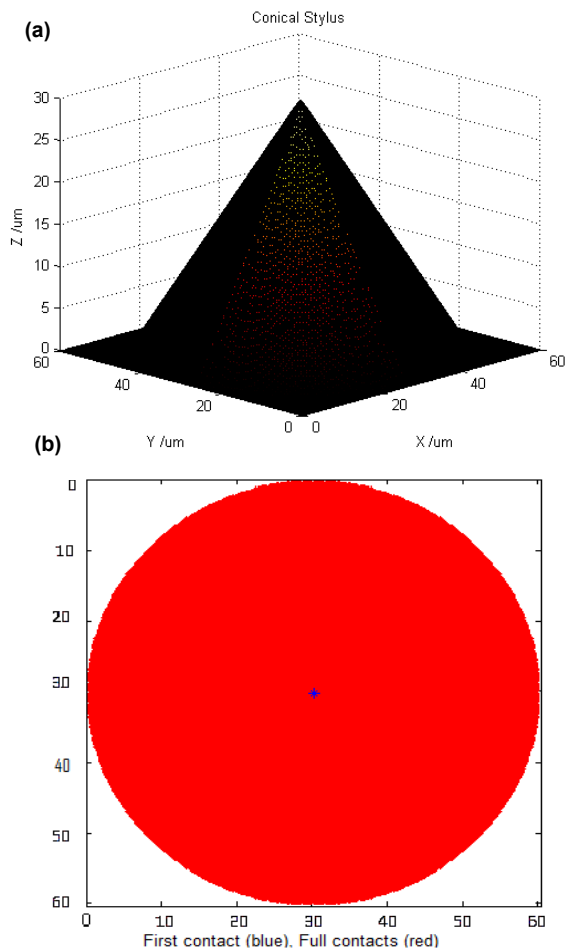


Fig. 1: (a) Conical shape 60x60 μm (b) Map of contact points with 149 nm threshold

As a very simple illustration of the technique, figure 1 shows an ideal conical stylus and a map indicating its contact against an ideal flat. The initial geometric contact is a single point. With a threshold, all surface points sufficiently close to the tip are considered to be in contact: a large value generates a symmetrical patch of contact. Here, we get effectively the same image as from an ideal indenter into a plastic material.

Figure 2 shows just the tip of the 10X10 μm conical stylus and contact with a flat using a much smaller threshold. The basic behavior is the same, but resolution limits within the geometry become apparent.

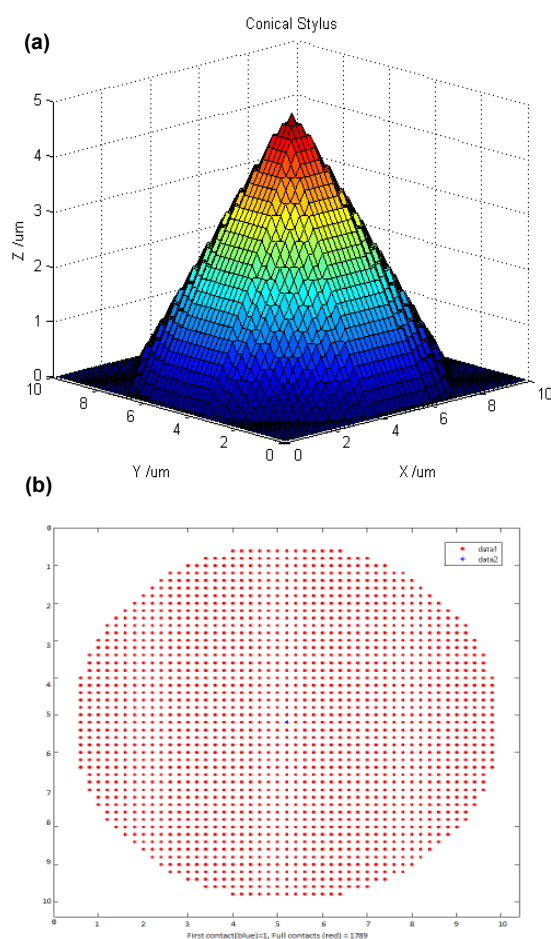


Fig. 2: (a) Conical shape Stylus 10x10 μm
(b) Map of contact points with 23nm threshold

Figure 3 shows a 20x20 μm small map (64 by 64 points on a 0.2 μm sample grid) taken by atomic force microscopy (AFM) from a typical ground steel surface. The fine detail has both sharply varying and plateau-like regions.

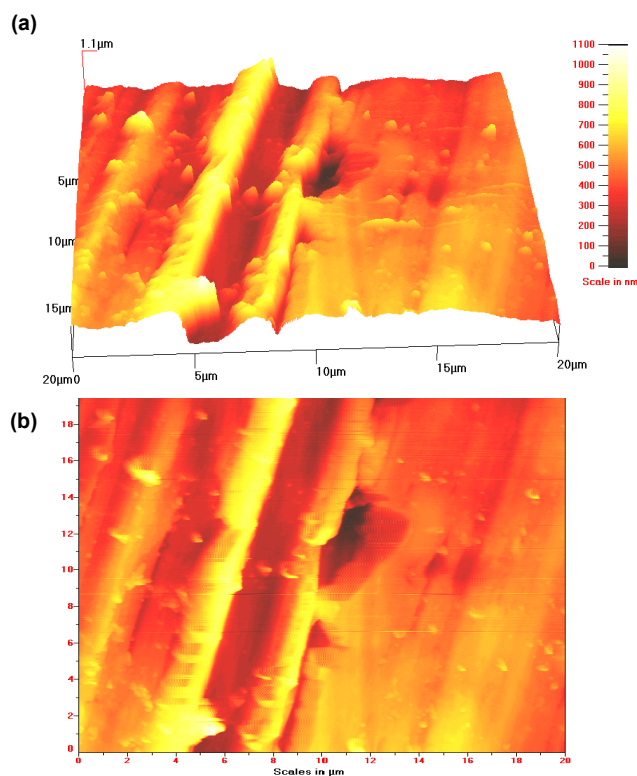


Fig. 3: (a) 3D AFM image (b) 2D AFM image

Figure 4 shows the stylus in (fig 2a) contacting local features of ground surface (fig 3a). Again using heavy threshold for clarity, key observations (fig 4b) are that the initial contact is not central and the patch grows asymmetrically, eventually picking out the major structures of the surface.

The total local height variation over this 10 μm by 10 μm is about 1000 nm, so thresholding to a few percent of height suggests real working simulations should be perhaps 10-20 nm.

To measure such a surface we might expect to use an instrument range of perhaps 5000 nm, with inherent noise of up to 1% full scale. Thus thresholding to a few tens of nanometers is also appropriate for modeling noise sensitivity.

For the particular condition illustrated, the map remains essentially the single point (in blue at Figure 4b) for thresholds as high as 40 nm; there is a single, real peak that is likely to dominate local contact behavior at the scale of this tip.

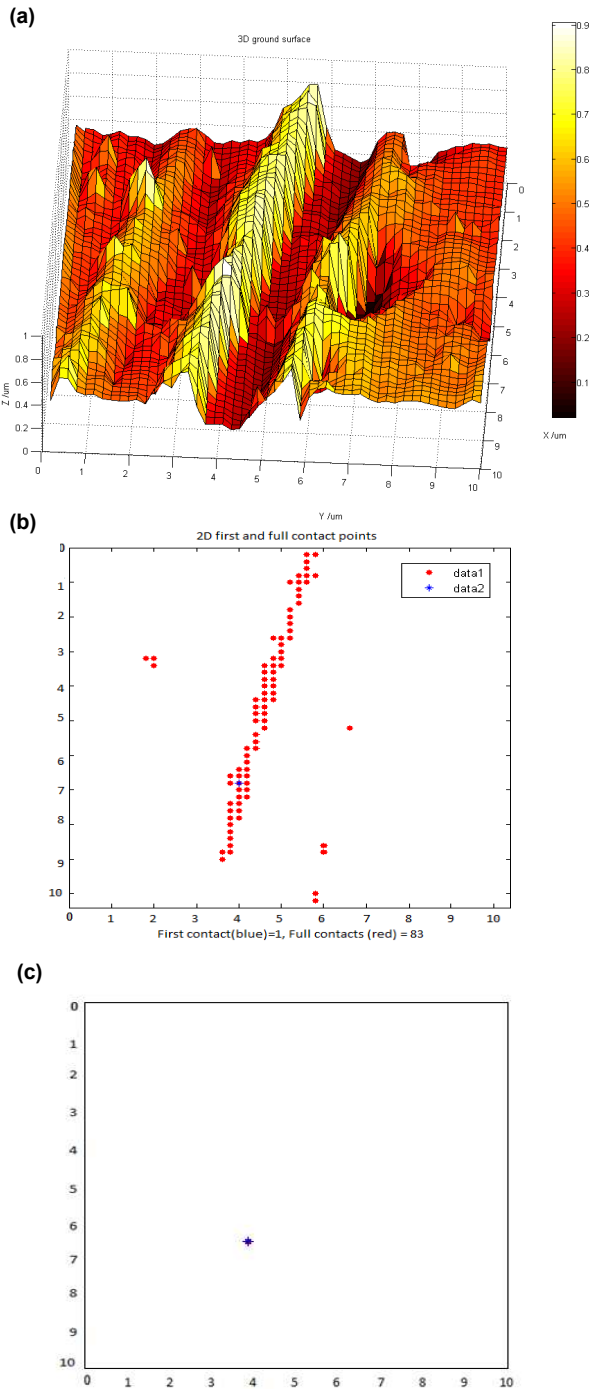


Fig. 4: (a) 3D Ground surface 10X10µm (b) Map of contact with 149 nm threshold (c) Map of contact with 40 nm threshold.

Figure 4c shows the threshold up to 40 nm leaves single point, which is the first contact. Comparing fig.4, and fig.5 (after moving the surface across and putting the tip in the middle), by applying heavy threshold, pattern is very

similar. It can be concluded that here only dominant peak and not noise sensitive. In fig. 4, the lateral error is about 1.5 µm. It also showed that here (tip radius fairly large) little change in reported contact under local movements. (i.e. not much different to flat stylus in actual behavior)

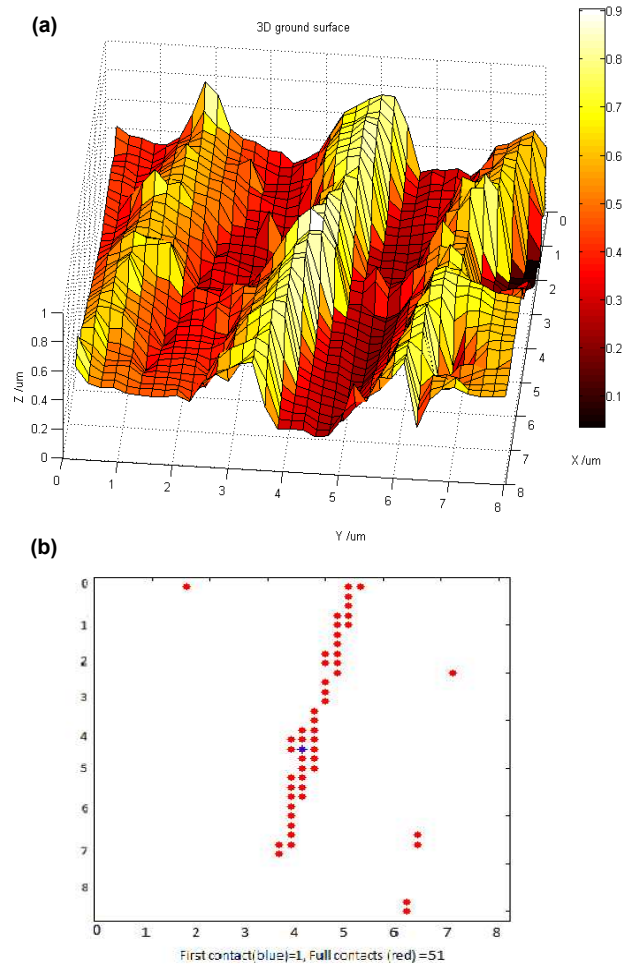


Fig. 5: (a) 3D Ground surface 8X8µm with tip at the centre. (b) Map of contact with 149 nm threshold.

CONCLUDING COMMENTS

Brief details of a new stylus contact simulation involving study of contact details on the tip itself and the use of a threshold function to test for sensitivity to noise and other functionally less significant features having undue influence on the predictions. The process has been illustrated using mainly unrealistically large thresholds for clarity. Initial evidence shows that examination of the contact patterns as threshold increases can identify the intensity with which different asperity regions interact with the stylus.

The method is therefore judged that the approach is worth further, more detailed study. Clearly, uncertainty predictions for real instruments will use small thresholds representative of noise levels and elastic deflections. There will then be a statistical accumulation of individually small effects across the surface, for which summary statistics are under development.

ACKNOWLEDGMENT

I would like to take this opportunity to express my sincere thanks to my home University in Saudi Arabia (Umm Al-Qura University) for funding my PhD study at Warwick University. I would also like to express my honest gratitude to the Royal Embassy of Saudi Arabia in London (Cultural Bureau) for their unlimited support and encouragement.

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SIMPLE CALIBRATION ARTIFACT FOR USE ON ROUNDNESS TESTERS

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ABSTRACT

When pursuing perfection, grinder operators need confidence that their measurement results are good to the microinch level, and they need a fast way to ensure this. A special concern is false readings due to lost motion in the structural loop. Electronic indicators can be calibrated off-line but that addresses only part of the problem. We describe a way to generate an artifact which emulates a workpiece having a known form error so that the entire roundness tester can be qualified.



FIGURE 1. Artifact before inducing distortion.

MOTIVATION

Air bearing spindles require components with form errors under 10 microinches. This requires at-the-machine metrology capable of resolving a microinch. In practice, resolution is lost when any portion of the structural loop has hysteresis. Examples include loose indicator tips, internal friction in the indicator, parts rocking on the chuck, loose joints in the support structure, and dirt under the surface gage. We make every effort to eliminate these problems but they have a way of creeping back. As a result, a conscientious operator tends to worry and when you worry you go slower.

What is desired is a way to prove at a moment's notice that the gaging is reliable. Our purpose-built roundness testers are mostly used to measure circular flatness so we developed this particular tool; a similar method could generate a comparable roundness calibration artifact.

DESIGN

We produce the form error by straining the artifact, lapping it flat, releasing the stress, and then calibrating the shape and size of the desired form. Stress is provided by screw tension and screw spacing sets the pattern. The calibrated zone is a narrow land so that there is less uncertainty about where measurements are to be made. Lapping is done in a circular pattern to minimize the effect of minute scratches.



FIGURE 2. Artifact with screws torqued to this backing plate, ready for lapping.

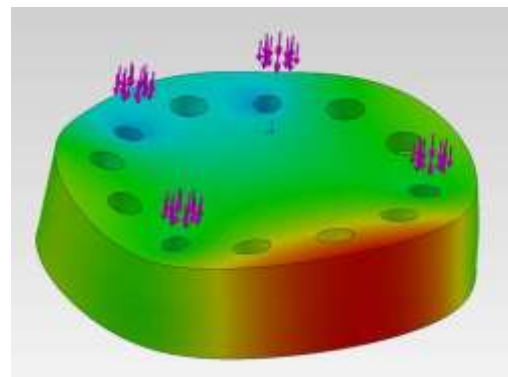


FIGURE 3. FEA of the artifact with screws torqued.

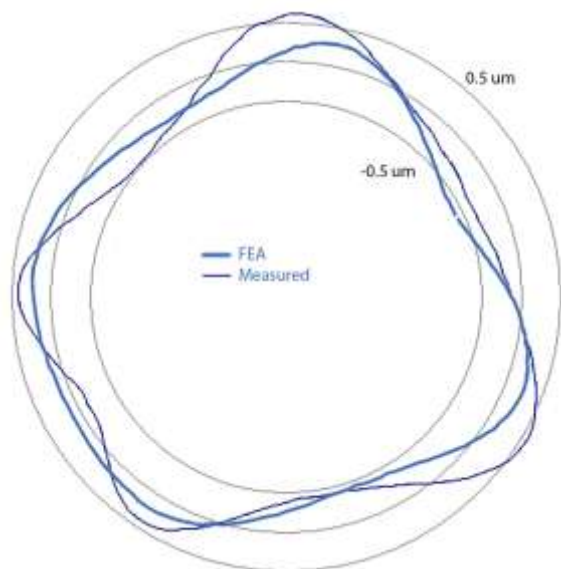


FIGURE 4. Comparison of FEA with measured results.



FIGURE 5. Lapping the artifact.

CALIBRATION

A Moore Measuring Machine provides a robust structural loop from the displacement indicator to the artifact, an encoder-equipped air bearing spindle supplies the reference axis of rotation, and a diamond-tipped air bearing cap gage contacts the artifact. Custom software produces numerical values, a polar plot, and a frequency display. If desired, the artifact can be verified at any facility with a roundness tester.

We use 50 Undulation Per Revolution filter because the errors we are looking for do not affect the surface finish, but rather are usually two-lobe from rocking on the grinder or three-lobe from previous chuckings.



FIGURE 6. Flatness artifact calibrated in roundness tester with Lion air bearing C-LVDT.

USE

This is a workaday tool, available for use at any time at any of our many roundness testers.

Since the artifact is similar to a typical workpiece it is a simple matter to install it and compare the shape and size of the form error with the calibration report. And, since the at-the-machine display uses the same software as the calibrator, examining the two images is like a blink comparator.



FIGURE 7. Calibrating a roundness tester on the shop floor.

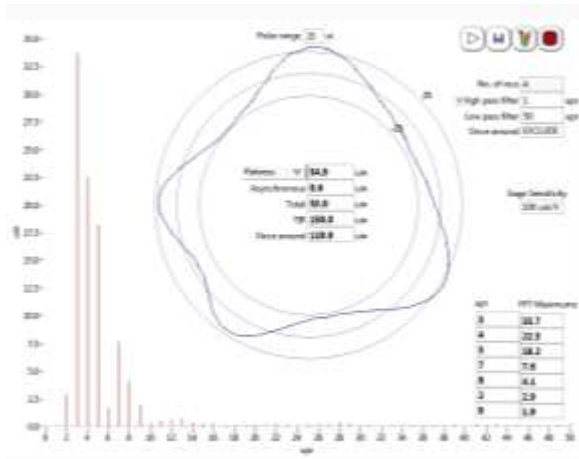


FIGURE 8. Circular flatness calibration result using roundness tester with Lion air bearing C-LVDT.

CONCLUSIONS

An artifact with a known form error is useful for confirming that at-the-machine metrology is trustworthy to the microinch level. A practical method is described for producing and calibrating a generic artifact.

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DESIGN AND CONSTRUCTION OF A PRECISION ROTATING MAGNETOMETER

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ABSTRACT

The next generation NIST watt balance will feature a large samarium-cobalt permanent magnet (860 kg) as the source of its magnetic field. The magnet, currently in fabrication, must generate a field between a pair of pole pieces of 0.55 Tesla uniform along a prescribed axis to 100 ppm (or ± 55 micro-Tesla (μT)). Such uniformity is difficult to verify using standard instrumentation. Here we describe the design and construction of a new magnetometer capable of precise measurement of the field strength via monitoring the magnitude of the voltage induced in a rotating coil. We discuss the principle and main elements of this sensor, the methods devised to translate and center it within the air gap between the pole pieces, and an electro-optical interface for digitizing and communicating the raw analog signals from its rotating magnetic and capacitive sensors. Results from a prototype device are presented.

BACKGROUND

The next generation NIST watt balance (NIST-4) has a magnetic circuit comprised of two SmCo permanent magnets housed in mild steel yokes whose pole faces define an air gap. The radial distribution of the flux density B inside the air gap is strictly governed, and can be manipulated, by the pole face geometries [1]-[2].

The air gap inside the magnet assembly measures 3 cm wide and 8 cm long, which can be externally accessed through 12 entrance holes located on the top and bottom surfaces of the outer yoke. The magnetic field strength inside the air gap is designed to be 0.55 T and should deviate by less than 100 ppm (or ± 55 μT , comparable to the Earth's surface magnetic field) at all points on the central axis of the air gap, as depicted in Figure 1.

The magnetic system implemented by the current watt balance has an outstanding field uniformity that deviates in a systematic fashion

by as little as 100 ppm about a mean value; a benchmark for what the new magnet should reach or surpass. This existing magnetic field has been functionally characterized using the large moving coil that is part of the apparatus. The voltage induced in this coil is accurately tracked as a function of its motions, in effect mapping the field profile during each experiment (see Steiner et al [3]).

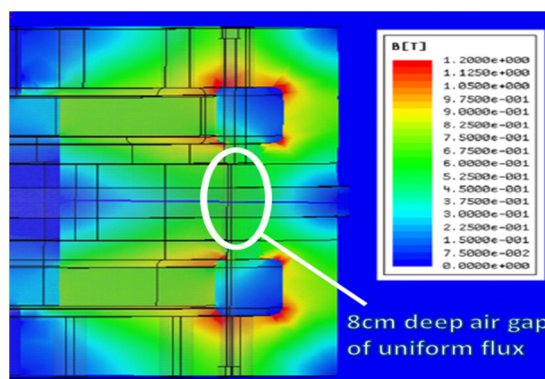


FIGURE 1. Magnetic flux density distribution inside air gap [3]. Field strength must be 0.55 T ± 55 μT at the central axis of the air gap in order to function as the watt balance's principal component.

For the new magnet, the expectation is that adherence to geometric tolerances (approximately 50 μm) in the machining of the components will produce the desired magnetic properties. However, the required geometric precision is likely difficult to achieve in the overall assembly, and it is anticipated that post assembly machining of the yoke, or other such modifications, will be required in order to compensate the magnetic circuit [4]. This requires a device that can measure the field at the absolute center of the air gap. Ideally, this device would be a moving coil, as in the present balance, but such a solution would require a suspension and position sensors that would be

too complex to implement at the factory where the magnet is being assembled.

Alternatively, magnetic field strength could be mapped using Hall probes. Typical Hall sensors can measure fields on the order of 0.55 T, but readout precision is only on the order of parts per thousand, which is inadequate for this application. In principle, the dynamic range can be extended, but the output is also dependent on orientation with respect to the field, so that errors in orientation might be misinterpreted as variations in the field. Thus, rather than trying to solve the problems associated with Hall probes, a novel sensing strategy based on a different type of moving coil was pursued, as we describe in the remainder of this paper.

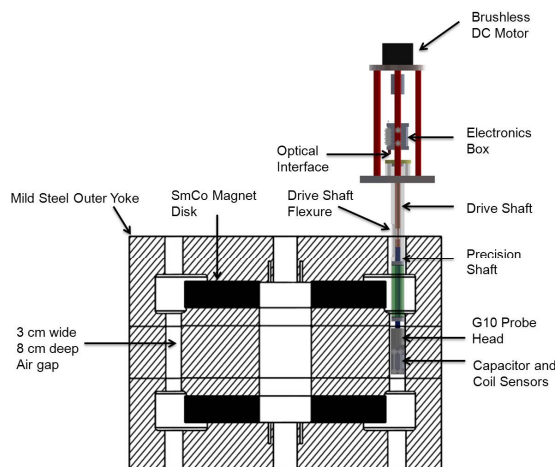


FIGURE 2. Cross sectional view of the SmCo permanent magnet assembly. The 2.54 cm diameter rotating magnetometer (without stand) is inserted into the 3 cm wide air gap.

PRINCIPLE OF OPERATION

To accurately measure the magnetic field, a unique instrument has been designed and constructed, represented in Figure 2. A brushless DC motor spins a 50.5 cm long drive shaft attached to a probe head. Consisting of a 60 turn copper coil, the probe head rotates at a selectable RPM to generate a sinusoidal induced voltage caused by the fixed magnetic field.

Equation (1) describes the relationship between the induced voltage V and the magnetic field B .

$$B = \frac{V}{NA\omega \sin(\omega t)} \quad (1)$$

Where N is the number of turns in the coil, A is the area of the coil, and ω is the angular velocity. The radial center of the air gap is significant in that it will be the location where the large moving coil resides. The probe must first locate this position and then measure the magnetic flux locally. This is done using a set of differential capacitors that sandwich the probe coil and rotate coaxially. Each probe plate couples with the nearest magnetic pole face, creating a differential capacitance reading that indicates probe position. Since the probe head rotates, this signal is modulated at the rotation frequency. The center location is found once the capacitor pairs have a constant differential reading.

MECHANICAL DESIGN

The initial iteration of the probe included only one single drive shaft connecting the motor to the probe head. Straightness imperfections and runout of the shaft were amplified as the distance from the motor increased. To remedy this issue, the single shaft was severed into two sections: a main drive shaft and a short segment precision shaft, represented in Figure 3. Two flexures (0.20 cm dia x 1.50 cm long) added to the drive shaft compensate for eccentricity and remove the majority of the runout through minimal load bending. This engineering practice ultimately decoupled the vibration source from the precision shaft.

The ceramic coated precision shaft is radially bound by two bearings and connects the drive shaft to the probe head, allowing the probe head to rotate true to its axis, depicted in Figure 3. This design reduced the runout by five orders of magnitude and produced a final runout reading of only 2.5 micrometers.

To position the probe in the air gap, a 5 DOF (degree of freedom) motion system is necessary. A fine pitch translation stage moves the system horizontally along the radial axis, necessary to adjust the capacitor feedback. A vertical translation stage allows the probe to shift in the up/down direction to scan the whole depth of the air gap. Rotation of the probe is generated by a brushless DC motor while the tip and tilt of the frame is adjusted by stand screws arranged in a three point contact formation.

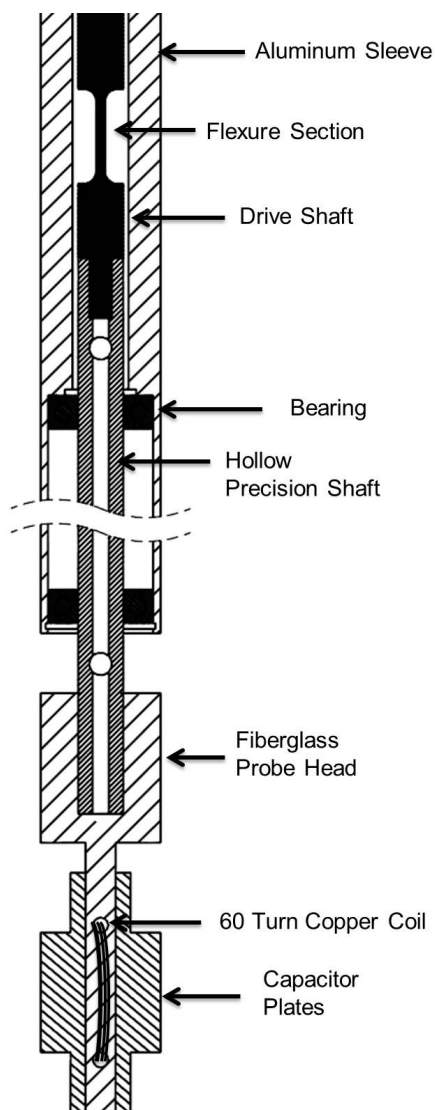


FIGURE 3. Cross sectional view of the probe assembly (bottom half). Only one flexure is shown.

ELECTRICAL DESIGN

Four copper wires (two for the rotating coil and two for the capacitors) deliver the analog signals to the electronics and are read in an alternating fashion. These wires feed directly through the hollow precision shaft to avoid interference with the bearings and are guided along the surface of the drive shaft. A thin burrow on the drive shaft located at the top bearing allows the wires to bypass the bearing bore with minimal intrusion.

The wires relay the dual channel, coil and capacitor, data to the signal conditioning unit, located on the rotating shaft below the motor.

This location was chosen to avoid interference of the magnetic field with the integrated circuits. We have built a prototype of the signal conditioning unit using a Microchip* PIC18F648 microcontroller, an MS3110P differential capacitance readout chip, and other passive components that encode the data from the two channels into serial data. The signal conditioning unit is fixed on the shaft and rotates in sync with the probe head. Data from the rotating system is transmitted via infrared LEDs to a stationary receiver using phototransistors. The signals from the receiver are converted to TTL signals and recorded via a National Instruments data acquisition board, NI BNC-2090A, to a labVIEW program for analysis.

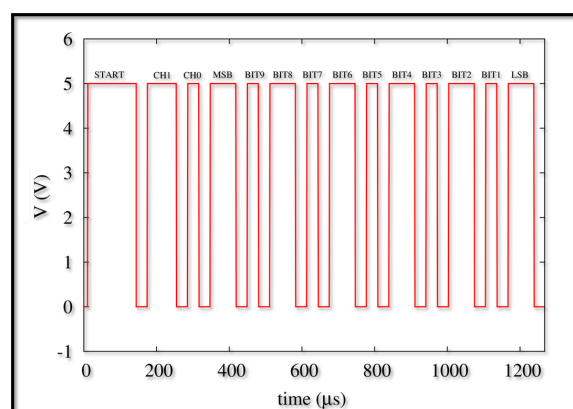


FIGURE 4. Transmission sequence encoding for data transfer to stationary receiver using the prototype signal conditioning unit. The prototype unit demonstrates the proof of principle with a 10-bit resolution. Pulse widths are not to scale.

The microcontroller reads each channel with an ADC of 10-bit resolution and encodes the corresponding digitalized data into a sequence of pulses with a delay of 1 μ s, as shown in Figure 4. The encoding scheme consists of a 155 μ s start-bit pulse which indicates data transmission. Subsequently following the start-bit are 2-bit pulses indicating channel (coil or capacitor) and 10-bit pulses indicating the actual data for the specified channel. The channel and data pulses consist of 80 μ s and 30 μ s pulses, with an 80 μ s pulse indicating a zero-bit and the 30 μ s pulse indicating a 1-bit.

DESIGN IMPROVEMENTS

In the first iteration of the probe, a slip ring was used to carry the analog signals from the probe head to the data acquisition system. The slip ring was rated for a maximum rotation rate of

300 rpm. In addition, the slip ring introduced noise to the voltage sinusoid due to unreliable connections. The new digital-infrared interface improved both limitations. With the infrared interface, reliable communication would be possible even with angular velocities that would surpass the mechanical failure point of the system. The noise, caused by resistance variations, is entirely avoided. Figures 5 and 6 depict waveforms from slip ring and optical interface respectively.

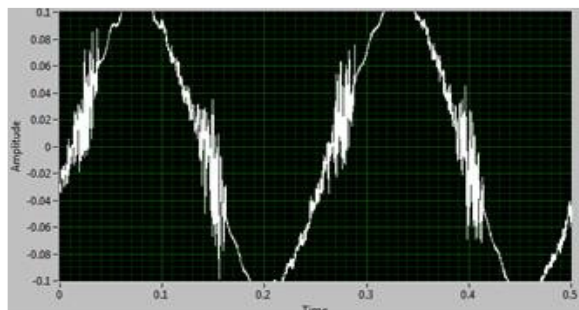


FIGURE 5. High noise levels from slip ring data attributed to poor internal connections of slip ring.

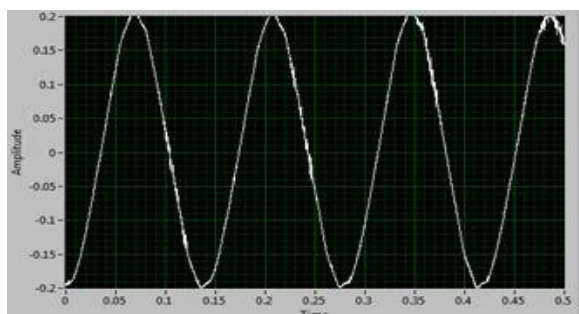


FIGURE 5. Low noise levels from optical data relay.

CONCLUSIONS

Whitworth's adage, "we can only make as well as we can measure", still rings true. Through good engineering practices, such as vibrational decoupling and wireless interfacing, a functioning magnetometer with centering capabilities has been designed and constructed. We have started testing the probe to verify its design specifications. The instrument will be used to measure the magnetic field generated once the permanent magnet is fully manufactured.

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DISCLAIMER

* Certain commercial equipment, instruments, or materials are identified in this paper in order to specify the experimental procedure adequately. Such identification is not intended to imply recommendation or endorsement by the National Institute of Standards and Technology, nor is it intended to imply that the materials or equipment identified are necessarily the best available for the purpose.

AFM LINEWIDTH MEASUREMENT EMPLOYING 3-D IMAGE REGISTRATION ON MULTI-VIEW SCANS

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ABSTRACT

This article presents an effective 3-D data registration algorithm for accurate linewidth measurement establishment. In the proposed methodology, a grating target is measured by an AFM (atomic force microscopy) from two measuring directions, to obtain optimal 3-D measured point clouds. The target is oriented at two unknown coordinate systems for the measurement of the step height from two different approaching directions by AFM. To merge the measured results accurately, the effective 3-D data registration has to be developed, so the point clouds obtained from the two orientation of the part can be merged seamlessly. With the calibrated linewidth and measuring parameters, the measurement traceability of the standard target and standards can be further enhanced.

INTRODUCTION

Atomic force microscopy (AFM) [1-2] is a very high-resolution type of scanning probe microscopy, with demonstrated resolution on the order of fractions of a nanometer, more than 1000 times better than the optical diffraction limit. AFM provides a 3-D profile of the surface on a nanoscale, by measuring forces between a sharp probe (<10 nm) and surface at very short distance (0.2 ~ 10 nm probe-sample separation). The probe is supported on a flexible cantilever. The AFM tip "gently" touches the surface and records the small force between the probe and the surface.

This work has employed an AFM system to measure the 3-D surface of nano-objects. However, the AFM system cannot measure the 3D global surface by one time due to a tip interference effect. To overcome this problem, the object surface is scanned by several times at various viewpoints. These scans are then align together to reconstruct the global surface.

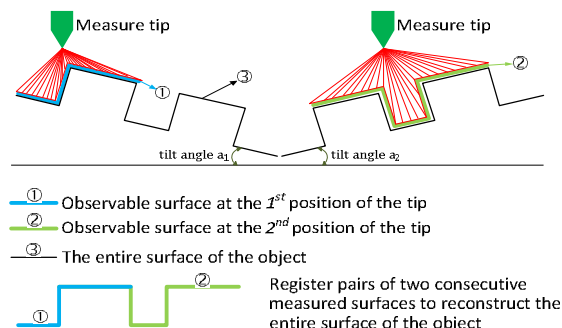


FIGURE 1. The problem for 3-D surface reconstruction for measured 3-D profile at various view scans.

The solution to the frame alignment problem strongly depends on the data being used. For the 3-D data, the *Iterative Closest Point* (ICP) algorithm [3] and variants thereof, such as the Trimmed ICP approach (TrICP) [4], the Picky ICP algorithm [5], the Iterative Closest Reciprocal Point algorithm (ICRP) [6], the X84 rejection rule [7] are popular techniques. The ICP algorithm iterates between associating each point in one time frame to the closest point in the other frame and computing the rigid transformation that minimizes distance between the point pairs.

METHODOLOGY

Data acquisition

3-D profiles of the surface are acquired using an AFM system at various viewpoints for the same measure object.



FIGURE 2. Atomic force microscope system employed for data acquisition.

Data representation

A collection of 3-D points allows having a discrete representation of the 3-D profile of the surfaces. A point p_i , described by $\{x_i, y_i, z_i\}$, is a primitive representation unit in a three-dimensional (3-D) Euclidean space. By definition, a collection of 3-D points is referred as a *cloud* structure $\mathcal{P}$. Every point p_i is given with respect to a fixed coordinate system. In most of the cases, this coordinate system has its origin at the sensing device used to acquire the data. For this reason, we can understand a point p_i as the distance on the three defined coordinate axes between the acquisition viewpoint and the surface from where the point has been sampled on.

Global 3-D surface alignment

Each area of the surface is scanned by two different viewpoints. These two scans are then applied to the registration algorithm to obtain the rigid transformation matrix.

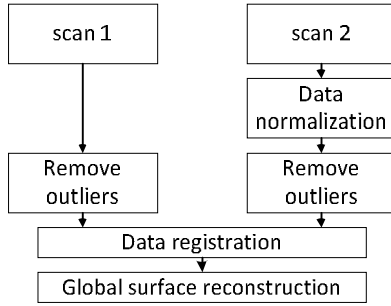


FIGURE 3. Flowchart of the registration algorithm for global 3-D surface reconstruction.

Data normalization

The second scan is rotated by 180 degree, therefore it has to be normalized first. To normalize, the data is rotated around z-axis by 180 degrees, and then translated by (T_x, T_y) .

The normalization matrix is given by

$$RT = \begin{bmatrix} -1 & 0 & 0 & 2000 \\ 0 & -1 & 0 & 2000 \\ 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 1 \end{bmatrix} \quad (1)$$

Outlier removal

The taken data from the AFM system may possess measurement errors that create sparse outliers. Such points can lead to errors while estimating rigid transformation in the step of alignment two scans. Moreover, removing such points will contribute to reduce processing time as it decreases the remaining point's number. The method used to remove outliers is based on performing statistical analysis on each point's neighbors. Assuming a Gaussian distribution, with a mean and a standard deviation, all points

having their mean distances outside a certain interval are considered as outliers and are removed from the dataset.

Mean distance from each point in the cloud to all its neighbors is calculated as follows.

$$\mu_i = \frac{1}{N} \sum_{k=1}^N \|p_k - q_i\| \quad (2)$$

Where

- $p_k, k = 1 \sim N$, are neighbors at the query point q_i in the cloud.
- N is number of neighbors at the query point q_i in the cloud.
- $\|p_k - q_i\|$ is Euclidean distance between point q_i and point p_k .

Data registration

Let P_s is the source point set, and P_t is the target point set.

$$\begin{aligned} P_s &= \{\tilde{p}_s\} \\ P_t &= \{\tilde{p}_t\} \end{aligned} \quad (3)$$

P_s consists of good 3-D points in the first dataset, while P_t consists of good 3-D points in the second dataset. Of course, number of features in P_s : N_s and P_t : N_t is equal. $\{\tilde{p}_{si}\}$ and $\{\tilde{p}_{ti}\}$, where $i = 1 \div N_s$, are corresponding together pair-by-pair. The rigid transformation, TR^k , and mean square error, MSE_k , are then calculated at the k^{th} iteration. MSE_k is compared with a threshold so that can achieve the best transformation.

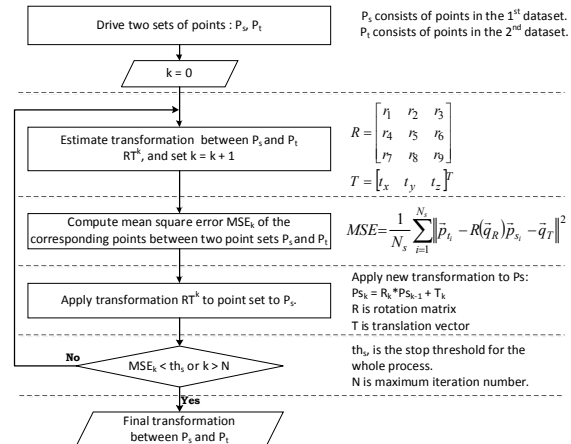


FIGURE 4. Flowchart of the rigid transformation estimate algorithm.

Estimate initial transformation

The "center of mass" of these point sets is given by

$$\begin{aligned}\bar{\mu}_s &= \frac{1}{N_s} \sum_{i=1}^{N_s} \bar{p}_s \quad \text{where } N_s = N_t \\ \bar{\mu}_t &= \frac{1}{N_t} \sum_{i=1}^{N_t} \bar{p}_t\end{aligned}\quad (4)$$

Determine cross-covariance matrix between P_s and P_t :

$$\Sigma_{src\_tgt} = \frac{1}{N_s} \sum_{i=1}^{N_s} [(\bar{p}_{s_i} - \bar{\mu}_s)(\bar{p}_{t_i} - \bar{\mu}_t)^T] \quad (5)$$

A symmetric 4 x 4 matrix is the formed as follows:

$$Q(\Sigma_{src\_tgt}) = \begin{bmatrix} tr(\Sigma_{src\_tgt}) & \Delta^T \\ \Delta & \Sigma_{src\_tgt} + \Sigma_{src\_tgt}^T - tr(\Sigma_{src\_tgt})I_3 \end{bmatrix}$$

where

$$\Delta = [A_{23} \quad A_{31} \quad A_{12}]^T \text{ and } A_{ij} = (\Sigma_{src\_tgt} - \Sigma_{src\_tgt}^T)_{ij}$$

$$\text{and } I_3 = \begin{bmatrix} 1 & 0 & 0 \\ 0 & 1 & 0 \\ 0 & 0 & 1 \end{bmatrix} : 3 \times 3 \text{ identity matrix}$$

The unit eigenvector corresponding to the maximum eigenvalue of the matrix $Q(\Sigma_{src\_tgt})$ is given by

$$\bar{q}_R = [q_0 \quad q_1 \quad q_2 \quad q_3]^T \quad (7)$$

The optimal rotation matrix between source and target point set is estimated by

$$R = \begin{bmatrix} q_0^2 + q_1^2 - q_2^2 - q_3^2 & 2(q_1q_2 - q_0q_3) & 2(q_1q_3 + q_0q_2) \\ 2(q_1q_2 + q_0q_3) & q_0^2 + q_2^2 - q_1^2 - q_3^2 & 2(q_2q_3 - q_0q_1) \\ 2(q_1q_3 - q_0q_2) & 2(q_2q_3 + q_0q_1) & q_0^2 + q_3^2 - q_1^2 - q_2^2 \end{bmatrix} \quad (8)$$

The optimal translation vector between source and target point set is given by

$$\bar{q}_T = \bar{\mu}_t - R(\bar{q}_R)\bar{\mu}_s \quad (9)$$

Where

$$\bar{q}_T = [q_4 \quad q_5 \quad q_6]^T.$$

The iteration is initialized by setting $\bar{q}_{T_0} = [0 \quad 0 \quad 0]^T$.

The mean square objective function

The mean square objective function to be minimized is

$$MSE_k = \frac{1}{N_s} \sum_{i=1}^{N_s} \|\bar{p}_{t_i} - R(\bar{q}_R)\bar{p}_{s_i} - \bar{q}_T\|^2 \quad (10)$$

EXPERIMENTAL RESULTS AND ANALYSES

Data extraction

Using software NanoScope Analysis to extract the data as 512 x 512 depth values. A text file which contains depth values is generated, in which each depth value is stored in a line. There

are total 512x512 lines corresponding to 512x512 scanned depth values.

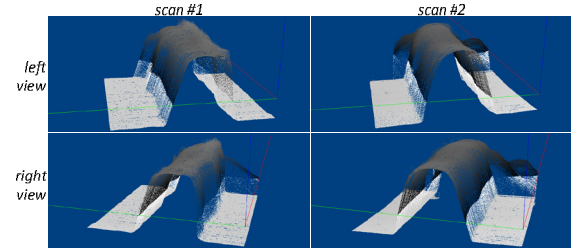


FIGURE 5. Clouds are formed from a pair of scans. Number of extracted points is 262,144 points for each cloud.

Resolution calculation for the scanned data

The scan size is 1.99 μm , each scan is mapped by a 512x512 2-D image. Consequently, the resolution for the scanned data is given by

$$\text{resolution} = \frac{2}{512} \times 10^3 (\text{nm}) \quad (11)$$

3-D points formation

According to depth values which are stored in the text file and the calculated resolution for the data, 3-D points are then formed as follows.

1. **for** (i = 0 ~ 511)
2. **for** (j = 0 ~ 511)
 - a. $z = \text{Read a line in the text file}$
 - b. $x = i * (2000 / 512)$
 - c. $y = j * (2000 / 512)$
 - d. new 3-D point = (x, y, z)
3. **end**

Filtering

Outliers are filtered based on the mean distance from each point in the cloud to all its neighbors. Number of the remaining points in the cloud is reduced significantly, leading to reduction of the processing time for the data registration step.

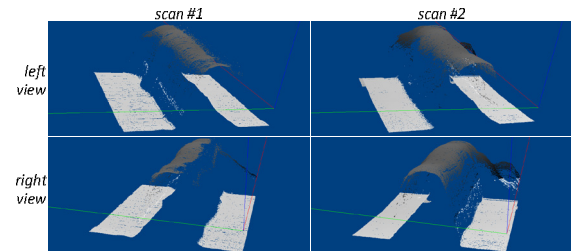


FIGURE 6. Filtered clouds are formed from the pair of scans. Number of retained points for the 1st scan and 2nd scan are 190,749 points and 211,653 points, respectively.

Outliers are considered as noises that contribute negatively to the data registration process. To guarantee the convergence and accuracy in alignment iterations, these outliers have been rejected and the processed data only retain

good points. In addition, number of points is reduced by 20% ~ 30%, this contributes to reduce processing time for the registration process. However, computation cost to reject outliers is expensive. The setting parameters for outlier removal, such as number of the neighbors, standard deviation to confirm a querying point is outlier or not, etc., is also a challenge and effects to the computation time and number of retained points for this step.

Data registration

To evaluate the accuracy of the alignment processing for two datasets, means square error (MSE) is calculate as follows.

$$MSE = \frac{1}{N} \sum_{i=1}^N \|p_i - q_i\| \quad (12)$$

Where p_i are points on the cross-section of the target dataset (*red line*), and q_i are points on the cross-section of the aligned dataset (*green line*); N is number of points within the range $[y_1, y_2]$; $\|p_i - q_i\|$ is Euclidean distance between point q_i and point p_i .

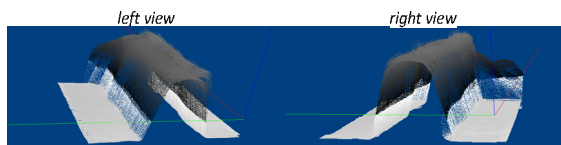


FIGURE 7. Aligned datasets for the pair of two scans.

According to robust regions on the surface, MSE is calculated to evaluate the accuracy for the data registration process. Values of MSE are in nanometer. Cross-sections are picked up at several of horizontal positions (*x-axis*). Line-widths are also calculated along each cross-section based on the specified height values.

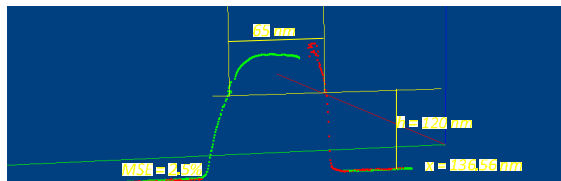


FIGURE 8. Evaluate the results of aligned datasets and line width measurement.

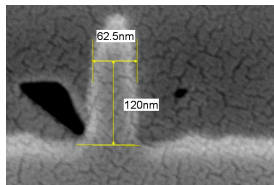


FIGURE 9. The SEM measurements which are used to compare with our results.

CONCLUSION

In this study, the ICP algorithm is used to stitch double images using an AFM instrument and the measurement results are compared with SEM. The major difference is caused by the diameter estimation of AFM tip. For this reason, in our future studies, we plan to develop a method to identify the profile of AFM tip.

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Reducing uncertainty of wettability measurements through controlled sessile droplet deposition: Current issues

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INTRODUCTION

Recent advances in technology such as adhesives used in micro-assemblies, composites in aerospace, advanced chemical filtration, and heat transfer epoxies in the semiconductor industry have incentivized a closer look at contact angle (CA) measurement from a precision metrological perspective.

More specifically as manufacturing scales shrink below the millimeter scale, interactions between components, and the processes they are subject to, are governed by a set of non-intuitive physical fundamentals [1]. At these scales capillary tensions become dominant over other traditionally significant factors such as gravity that play a key role in many of the established measurement methods. At these reduced scales both innovative new solutions and further refinement of existing CA measurement procedures are necessary if reduced uncertainties are to be achieved.

With these challenges in mind, the goal of this research is to advance the field of CA measurement to provide precision wettability values for localized areas on a surface. To do this it is necessary to reduce the dimensions of sessile drops below that recommended in current standards, and both identify and characterize optimal measurement processes.

THEORETICAL CONSIDERATIONS

A wetting system can be characterized by the wetting contact angle (CA) of a sessile drop deposited on a surface. In the absence of any other forces, the drop will assume a truncated spherical shape. The spherical shape is formed from the relatively weak cohesive forces that exist in liquids, compared to the liquid surface

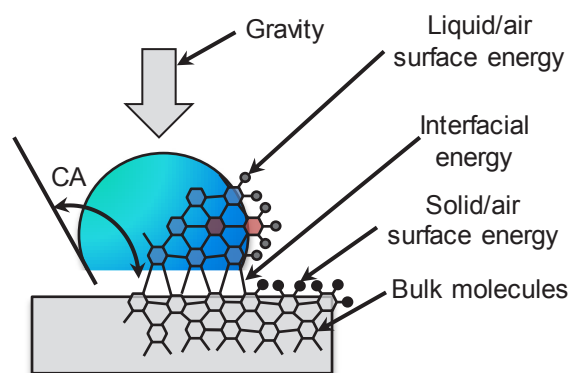


FIGURE 1. Depiction of influences acting on sessile drop and resulting definition of contact angle.

energy, γ_L , resulting in the droplet assuming a minimal energy state; the truncation of the sphere is a result of the cohesive forces acting between the bulk liquid and the bulk solid, γ_{SL} [2]. A depiction of a wetting system is shown in FIGURE 1.

A relationship between the cohesive and adhesive forces was found in 1805 when Thomas Young studied the correlation between surface energies, and the resulting angle of contact of liquids and solids [3]. This relationship is known as the Young's equation [4].

$$\gamma_S = \gamma_{SL} + \gamma_L \cos(\theta) \quad (1)$$

Where γ_S is the surface energy of the solid in air, γ_{SL} is the solid to liquid interfacial energy,

γ_L is the liquid in air surface energy, and θ is the wetting contact angle (CA).

To date, the Young's equation is being used in simplified models for estimating dispersion forces at interfaces, although for more precise measurement, more advanced techniques are used [5]. Likewise, it is typical to use higher level theoretical models that attempt to account for any of the numerous sensitivities in CA measurement. Thus it is important to note that the Young's equation is a model for an ideal measurement, which is one consisting of a chemically insoluble, atomically flat, and homogeneous substrate used with an equally homogeneous liquid, all tested in ideal air.

Notice the absence of gravity in the Young's equation. In theory, CA is independent of gravity; however for droplets measuring millimeters or larger, gravity has a measurable influence. Gravity as an influence can be accounted for using a Laplacian correction as a function of drop diameter [6] and is typically incorporated into CA curve fitting software. As droplet diameters reduce, the ratio of surface area to volume increases until surface tension forces dominate those of gravity. Fortunately for our project, the effects of gravity are less significant as the sessile drop diameters become sub-millimeter (a 1 mm diameter spherical droplet has a volume of 0.52 μL). The insignificance of gravity at these small volumes considerably simplifies analysis.

Perhaps the most challenging obstacle in CA measurement is dealing with hysteresis. This can be readily conceptualized with the use of the term 'pinning'. In general it is a term that is used to describe the movement, or lack thereof, of the triple line in the presence of a change in the system. This is a challenge because theory predicts that as a sessile drop is changed (through evaporation, adsorption or injection of new matter) the CA should remain constant facilitated by a moving triple line. In reality, for small disturbances, there is a greater probability that the triple line will remain in place and the contact angle will alter to maintain equilibrium.

In the past, attempts have been made to attribute hysteresis to changes occurring in a surface as that surface is wet and dewet. These ideas have since been replaced by the concepts of metastable states and heterogeneity [7,8].

The difference between what simple theories predict, and how this relates to real measurement procedures, is a significant influence in uncertainty estimation for measurements. Thus, it is informative to measure hysteresis and apply more advanced theories such as Zisman theory to correlate this information to the ideal CA. In an effort to enhance repeatability, Zisman focused on using well characterized droplet deposition processes to facilitate advancing contact angle measurements [7]. The advancing contact angle is the highest possible angle that can be achieved for a given wetting system. For sub-millimeter systems, it is typical for these two angles to be measured by dynamically changing the volume of sessile drops [9]. A challenge for this research project is that, with the reduction of sessile drop diameters below the 100 μm range, microfluidic effects become significant, in particular, large pressure gradients within the needle during extraction.

From the work of Gibbs published in 1884, hysteresis can be evaluated using a thermodynamic approach [8]. According to Gibbs, in an ideal measurement, there is a single minimum of free energy as a function of geometric contact angle. To differentiate, the geometric contact angle, θ_g , is the theoretical contact angle derived from geometry. In the case of a sessile drop, the geometric contact angle is a function of the base radius, r_b , and its volume, V given by [9]

$$\frac{3V}{\pi r_b^3} = \frac{2 - 3 \cos(\theta_g) + \cos^3(\theta_g)}{\sin^3(\theta_g)} \quad (2)$$

The Gibbs energy, G , of a system can then be found from

$$G = \gamma_L A_L + (\gamma_{SL} - \gamma_S) A_{SL} + \gamma_S A_{total} \quad (3)$$

Where A_L is the surface area of the liquid to air interface, A_{SL} is the surface area of the solid-liquid interface, and A_{total} is the total surface area of the solid substrate [9].

Gibbs also proposed the idea that heterogeneities within a system caused localized changes in the interfacial energies [8].

From these two equations it is apparent that there can be multiple solutions for the surface energies and areas for a given geometric contact angle. Consequently, there are also multiple contact angles corresponding to localized free energy minima. *FIGURE 2* shows the Gibbs energy plot for an ideal wetting system, and an exaggerated representation of the perturbations of the Gibbs energy for a wetting system with imperfections in the surface and heterogeneities in the materials. As shown, the ideal system has a single energy minimum, while the simulated measurement has many localized energy minima with 'energy barriers' between each. Each localized energy minima is referred to as a metastable state. One can see from this simulation that in an actual measurement the sessile drop could equilibrate to a metastable state that does not correspond to the ideal minima. The discrepancy in the measured CA from the ideal, or geometric contact angle, is one of the reasons that the term 'apparent contact angle' is commonly used.

From *FIGURE 2*, it is easier to see why it is suggested that the evaluation of the static contact angle yields little useful information for the characterization of a surface and that more information is provided by the advancing or receding contact angle. A major source of this uncertainty is attributed to the drop coming to rest at a metastable state as opposed to the ideal Gibbs energy minimum [10].

TYPICAL CONTACT ANGLE MEASURES

A review of literature and recommendations in available standards will yield many different perspectives of what a CA measurement should include.

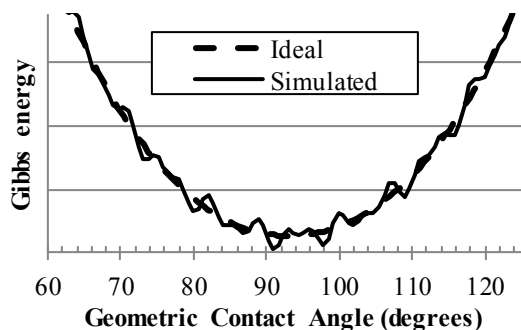


FIGURE 2. Plot of the Gibbs energy for a sessile drop of a given volume as a function of geometric contact angle. Both the energy of an ideal measurement and a simulated measurement with heterogeneities are shown.

As an overview, the following considerations are of relevance to our project and have been extracted from guidelines and journal papers. Because of both evaporation and time dependent spreading of the liquid onto the surface another measurement complication is the time between sessile drop deposition and measurement. For this reason it is advised to take the measurement within 10 seconds of deposition [7]. As the drop diameter approaches the scale of the heterogeneities in the testing materials, severe consequences to the CA can be evident. Johnson et. al. recommends that a droplet volumes be greater than 0.01 mL [7].

While standards exist, such as ASTM D7334-08 [10], the application-dependent nature of the CAM field requires such standards to be strategically vague to accommodate the variety of testing conditions. As specified there is a range of suggestions including, a voluntary cleaning procedure with soap and water, sessile drop volumes ranging from 5 μL to 20 μL with a volumetric control of $\pm 0.1 \mu\text{L}$, a testing temperature of $23 \pm 2^\circ\text{C}$, and testing environment with a relative humidity of less than 50% [10]. However, many researchers, this current project included, require material properties under conditions that do not fall in this range and may benefit greatly from a more comprehensive standard.

ADVANCING CONTACT ANGLE MEASUREMENT TO THE SUB-MILLIMETER SCALE

The motivation to reduce the scale to droplet diameters down to 10's of micrometer is driven by a few factors; to minimize the effects of gravity, to fit more drops in a given area allowing more measurements for averaging, to measure localized surface properties and to anticipate behavior of scaled interactions. To accomplish this task, a range of testing applications must be targeted and, within the range, a sensitivity analysis must be carried out to determine correlations between influences such as surface roughness, subsurface damage, and heterogeneity to variations in CA. For example the scale of inconsistencies likely to be present on semiconductor quality silicon wafers are presumably lower than that expected in a sample of commercial polymeric material. Therefore it may be acceptable to use picoliter drops on the silicon while limiting drops on the polymer to milliliters or larger. As sessile drops shrink, the information that is obtained from CA

measurement is a result of the information contained within that region; therefore if localized information is desired, then the scale of the sessile drops must match the scale of the areas of interest.

Considering the above discussions, an automated process is necessary. The goal of the automation is to reduce variability from the sessile drop deposition and measurement processes.

To facilitate this research, a fully automated 4-axis machine has been configured to be used in the sessile drop deposition process. A block diagram of the machine is shown in *FIGURE 3*. An automated xyz positioner is used to position an auto-syringe relative to the substrate. It is important to note that the process is controlled using positions, velocities, accelerations, and syringe plunger position calculated as a function of needle diameter, desired drop diameter or volume, and measurement times (all normalized to nominal final sessile drop parameters such as contact diameter and sessile drop height). A major goal is to develop deposition process coefficients and parameters to deposit a sessile drop with estimates for repeatabilities and combined uncertainties.

To date, experimental process parameters are being varied to assess repeatability. Thus far our group has successfully deposited 500 pL drops

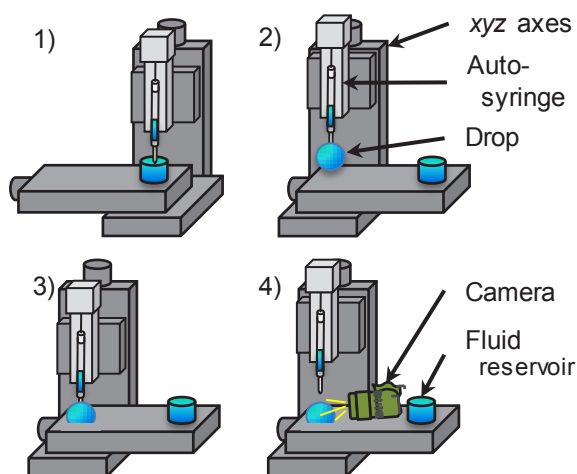


FIGURE 3. Block diagram of sessile drop deposition machine and deposition process. 1) drain and re-fill syringe, 2) move to drop extraction point and extract specified volume, 3) execute controlled deposition procedure, 4) measure drop.

having approximate dimensions of 150 μm in diameter when contact angles are near 70° with a 500 μm OD needle. It is anticipated that with further parameter refinement and the use of smaller diameter needles, the process will be capable of depositing in the 10's of picoliter range. With this capability, it is hoped to gain a quantitative insight to localized material properties within surface regions spanning 10's of micrometer range.

ACKNOWLEDGEMENTS

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OVERVIEW OF AXIS OF ROTATION CENTERING AND FIRST HARMONIC ERRORS FOR FIXED AND ROTATING SENSITIVE DIRECTION APPLICATIONS

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INTRODUCTION AND BACKGROUND

The measurement of axis of rotation errors is performed, in part, to estimate the uncertainty in machining and gauging applications¹. In many applications, the uncertainty of the axis of rotation (spindle) is a significant contributor to the overall system uncertainty and a cost driver.

Thus, to correctly establish overall system uncertainty, consistent measurement techniques and data reductions are essential in measuring axis of rotation errors². In these measurements, it is common to remove the 1st harmonic of measured spindle error. The removal of this component is justified as this component has been historically considering a positioning or centering error of an artifact or workpiece^{5,3}. This treatment is convenient as it speeds setup by reducing the time required to center the artifact. However, this approach reduces the reported error by the magnitude of the measured 1st harmonic and thus it is important to consider the validity or removing this component: i.e. data reduction must be consistent to ensure that components which contribute to workpiece error are not inadvertently removed.

Thus, it is important and timely to examine the validity of removing this error component and indicate circumstances where this may be inappropriate as has been suggested by Lu⁴.

The removal of the 1st harmonic as centering error is referenced in numerous publications, in industrial machinery user manuals³ and national standards.⁵ However, recently the validity of this has been questioned.⁴ In this review, the implicit and explicit definitions of centering error are stated and various cases of 1st harmonic error are then examined to determine if these should be treated as centering error or as spindle error motion. This is considered for both the cases of fixed and rotating sensitive direction applications.

Definition of Harmonic Content

A review of publications, standards, and industrial documents do not reveal an explicit mathematical definition of centering error and the 1st harmonic.

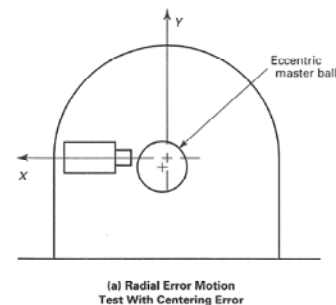


Figure 1 Centering error from standards⁵

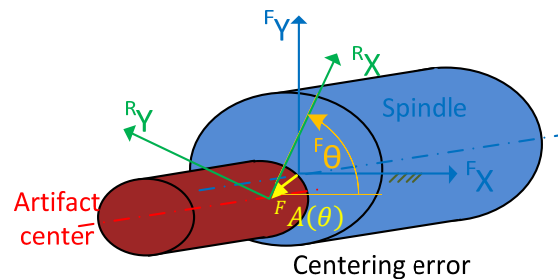


Figure 2 Artifact with centering error and labels

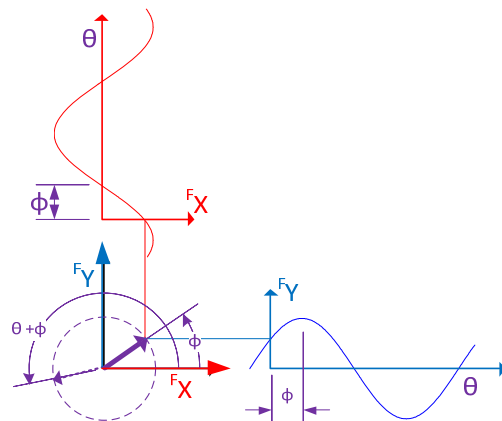


Figure 3 Centering error measurement

This omission presents a challenge in the standardization of signal processing and in the justification for removing 1st harmonic errors. Nevertheless, the required relationships can be inferred from standards documents such as that shown in Figure 1. For clarification, this figure is extended in Figure 2 with nomenclature.

Explicit Definition of Centering Error

Figure 1 and Figure 2 illustrate the 1st harmonic centering error, A, expressed in the fixed (F) frame where the superscripts F and R indicate fixed or rotating coordinate frames. Vector A includes xy components as a function of $^F\theta$

$$^FA(\theta) = \begin{bmatrix} X(^F\theta) \\ Y(^F\theta) \end{bmatrix}$$

It is noted that this is the error at a specific axial position and that—in general—five measurements are required to describe the axis error motion.^{1,2} The application of this definition of centering error gives the relationships between x and y—where an initial phase offset of ϕ —is allowed, and thus

$$X_1(^F\theta) = A_1 \cdot \cos(^F\theta - \phi_1)$$

$$Y_1(^F\theta) = A_1 \cdot \cos(^F\theta - \phi_1 - \pi/2) = \sin(^F\theta - \phi_1)$$

The generalized spindle motion in terms of Fourier coefficients is

$$X(^F\theta) = a_{x,0} + a_{x,1} \cdot \cos(^F\theta \cdot 1) + b_{x,1} \sin(^F\theta \cdot 1) + a_{x,2} \cdot \cos(^F\theta \cdot 2) + b_{x,2} \cdot \sin(^F\theta \cdot 2) + \dots$$

$$Y(^F\theta) = a_{y,0} + a_{y,1} \cdot \cos(^F\theta \cdot 1) + b_{y,1} \sin(^F\theta \cdot 1) + a_{y,2} \cdot \cos(^F\theta \cdot 2) + b_{y,2} \cdot \sin(^F\theta \cdot 2) + \dots$$

Thus, the explicit definition given here in Figure 1 and Figure 2—extended from the notion of a centering error as a 1st harmonic—shows that for this case, the response at y lags the response at x by 90° and thus

$$a_{y,1} = -b_{x,1} \quad \text{and} \quad b_{y,1} = a_{x,1}$$

Clearly, the Fourier coefficients of the x and y errors for the 1st harmonic are dependent based on our definition of centering error. This has been extended to higher harmonics above, but the coupling of these motions in x and y has not been—and cannot be—justified in general.

Fixed and Rotation Sensitive Directions

It has been shown that a duality exists between fixed and rotating sensitive direction applications.^{6,7,8} This duality appears to have been instrumental in results reported by Lu et. al.⁴ based on one interpretation of the authors' simulation results which appear to implicitly show mixed fixed and rotating sensitive directions results.

Based on this known duality, it is shown here that examining the axis of rotation harmonic errors from the FSD and rotating sensitive direction (RSD) perspectives gives better intuitive insight into the effects these errors on machining and metrology applications.

Illustration of Workpiece Errors due to 1st Harmonic Errors

The explicit definition given above is useful to illustrate the effect of centering errors. Figure 4 shows the effect of a 1st harmonic centering error. In this case, the xy components of the 1st harmonic are equal in the sensitive and nonsensitive direction (for fixed frame). This equality is implied in Figure 1.

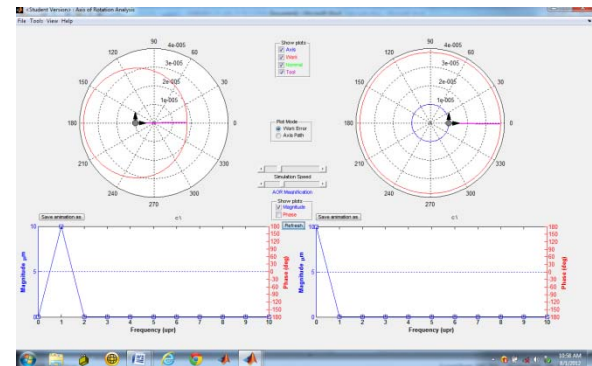


Figure 4 Classic definition of centering error:
Fixed Coordinate System vertical error = 10 µm
horizontal error = 10 µm

Figure 4 shows that a nonconcentricity error for the workpiece in the fixed sensitive direction case results as expected. It is seen from the upper left plot that the axis of rotation error is a single point that does not coincide with the desired workpiece center. The conventions for these plots are given below.

The two right hand plots in Figure 4 show the rotating sensitive direction case. In this case the “1st harmonic” axis of rotation error results in a 0th harmonic error (size) in the workpiece. Also, from the upper right hand corner plot, it is seen that the error in the sensitive direction (axis of rotation error) is also a 0th harmonic error. Thus, what was a 1st harmonic error for the FSD case is clearly a 0th harmonic error for the RSD case. The basis for this is shown by Miller⁶.

Figure 4 through Figure 7 illustrate the effect of a given axis of rotation error on both fixed and rotating sensitive direction applications. In each

graphic, the top left plot is the FSD polar plot and the bottom left plot is the frequency content of the workpiece error. The right two plots show the same information for the rotating sensitive direction case. In these figures, the path of the axis of rotation error is shown in blue, the workpiece error is shown in red and the tool (or probe) orientation is shown in purple. All results are expressed in the workpiece coordinate frame. For the case of a fixed sensitive direction application, the workpiece frame is the rotating frame. Thus, the plotted axis of rotation error is the error that sensors mounted on the tool or sensor would observe.

As is shown later, a better intuitive understanding of the effect of axis of rotation errors on application is gained by this approach.

Breaking the constraint between xy components of the 1st harmonic error

The error motions for retrograde 1st harmonic error as described by Lu,⁴ is shown in Figure 5. In these discussions, the validity of removing the 1st harmonic error as centering error was question and this issue is discussed here. In Figure 5, it is evident from the workpiece error that retrograde motion results in nonconcentricity error for FSD. From this result, it should be clear that it is always possible to reposition the workpiece to eliminate the 1st harmonic workpiece error. This is true since the axis of rotation error projected along the sensitive direction is always a 1st harmonic and the component in the nonsensitive direction is generally insignificant regardless its magnitude, sign or phase. Thus, the 1st harmonic error in the nonsensitive direction will remain but this does not significantly affect the workpiece error.

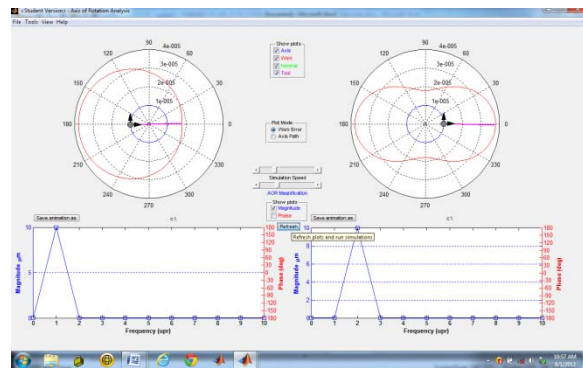


Figure 5 Retrograde motion: Fixed Coordinate System vertical error = -10 μm (retrograde) horizontal error = 10 μm

For the rotating sensitive direction, it is shown in Figure 5 that the retrograde 1st harmonic motion results in a 2nd harmonic (oval) workpiece error. This result is compared with a 1st harmonic centering error in the FSD case which—as was discussed above—results in a 0th harmonic error (size) for RSD case.

As discussed previously, this paper illustrates the usefulness of examining axis of rotation errors in both the rotating and fixed reference. While this approach is analogous to presenting all components of the axis of rotation error the method presented here allows continued use of the 1st harmonic notion of centering error and explicitly shows when this results applies and when it does not.

As the case of retrograde motion illustrates, it must be recognized that harmonic components in orthogonal directions are independent and—in general

$$a_{y,1} \neq -b_{x,1} \quad \text{and} \quad b_{y,1} \neq a_{x,1}$$

Clearly, this result is inconsistent with the stated notion of a first harmonic centering error where—based on our definition of centering error—the xy components due to such an error must be equal in magnitude. This inconsistency suggests that our definition of harmonic axis of rotation error can be improved.

Discussion of issues associated with identifying axis of rotation errors based on a single harmonic component

This issue is illustrated below using simulation for fixed and sensitive direction applications of axis of rotation error. In these examples, the effects on workpiece (or measurement) errors for both fixed and rotating sensitive direction applications are examined. In particular, the workpiece error resulting from a given axis of rotation error is simultaneously evaluated for both fixed and rotating sensitive directions—as before. This approach effectively illustrates the challenge associated with naming axes of rotation errors as 0th, 1st, 2nd, or higher harmonic errors without reference to the state (fixed or rotating) of the application. As illustrated by the examples in Figure 4 and Figure 5 as well as those below, it is clear that we need to specify the state of the application as fixed or rotating in order to justify the shorthand notion of specifying a single harmonic term to describe centering or other axis errors.

Figure 6 illustrates the results for retrograde motion when the components in the fixed xy direction are unequal. In this case, an additional 0th harmonic error is present in the rotating case where the magnitude of the 2nd harmonic is altered when compared to the previous case.

Figure 7 shows the results for forward motion where the components of the error in fixed xy are unequal. While the error for the FSD case remains a 1st harmonic or centering error and again a 0th (size) and a 2nd (oval) harmonic are present for the RSD case.

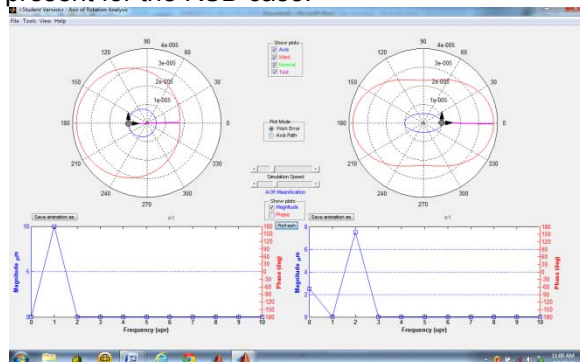


Figure 6 Retrograde unequal motion: Fixed Coordinate System vertical error = -5 μm (retrograde) horizontal error = 10 μm

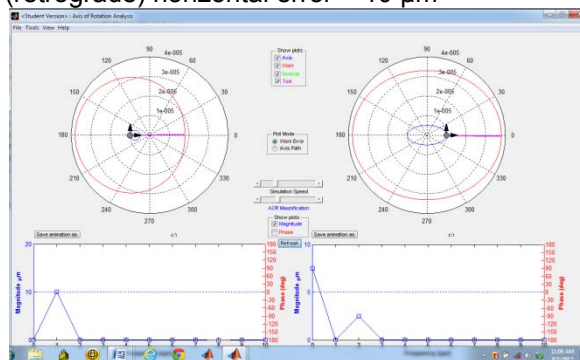


Figure 7 Forward motion unequal 1st harmonic error components: Fixed Coordinate System vertical error = 5 μm horizontal error = 10 μm

CONCLUSION:

This paper shows that the existing treatment of 1st harmonic errors in axis of rotation⁵ applications can be improved by specifying the state of the sensitive direction as fixed or rotation. Specifically, if the harmonic content of the axis of rotation error is expressed as the harmonic content of the sensitive direction: this is the approach taken historically for fixed sensitive direction application. In this approach, the frequency content of the axis of rotation error corresponds to the frequency content of workpiece or measurement error. Thus, the

reported spindle error is a function of the application and the tester must know the application in order to correctly state the spindle error. Clearly it is incorrect to remove unequal 1st harmonic errors for a spindle for a RSD, while for FSD it is acceptable.

Additionally, in the poster session it is shown that simulation of axis of rotation errors can improve the understanding of the effects of various harmonic axis of rotation errors. This treatment may help machine tool builders, users, and standards organizations to use consistent terminology. Specifically, it should be considered to name the 1st order (and higher) harmonic content for the given sensitive direction. Thus, in the examples above, the axis of rotation would be said to have a 1st harmonic error for a fixed sensitive direction. Under the same conditions, the axis of rotation would be said to have a 0th (size) and/or and 2nd (oval) harmonic error for RSD.

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CMM MEASUREMENT AND EVALUATION OF MODELS WITH TRIMMED FREE-FORM SURFACES

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INTRODUCTION

Free-form surfaces are used in product design, either to deliver certain functionality, or for aesthetic purposes. The realization of models with free-form surfaces involves metal removal processes whose output is either the desired product, or a die/mould cavity for the desired product. This process is affected by various sources of error, which cause the geometry of the manufactured components to deviate from the design intent, represented by the CAD model. It is, therefore, important to ensure that this deviation is quantified in order to be addressed in the design, and manufacturing processes.

Coordinate measuring machines (CMMs) are used to measure the deviation of the physical model from the CAD model. The inspection of free form surfaces is a challenging process due to the complexity, and shape irregularity of the physical model. The CMM inspection may be divided into three main activities: 1) inspection planning, 2) part measurement, and 3) tolerance analysis and verification.

A practical approach to overcome some issues which arise in the CMM inspection of trimmed free-form surfaces is proposed in this paper.

First, the free form surface features are measured along their isoparametric curves using the analogue scanning technology. A probe path planning algorithm is developed and implemented in this work. The purpose of this algorithm is to generate interference-free probe scanning path, which is used to guide the CMM navigation system to measure the physical model.

Second, a methodology for measurement data localization with the CAD model is proposed, and implemented.

Third, surface profile analysis is used to evaluate the measured surface with respect to the design requirement. Two techniques are used for profile analysis: 1) measurement points-to-model deviations, and 2) substitute geometry-to-model deviations. The latter is based on the reconstruction of the measured geometry using skinning of cross-section curves.

The work demonstrated in this article was carried-out using the Carl Zeiss VAST analogue scanning technology [3, 5].

INSPECTION PLANNING

Inspection planning consists of determining the locations where the physical model is to be measured, and the collision-free probe scanning path. Of particular interest to this work is the sampling of free-form surfaces. The physical model is measured along its isoparametric curves, which are distributed based on the surface geometry, and topology [1, 2]. The CAD model, and the sampling plan for a trimmed surface are illustrated in figures 1, and 2, respectively.

The sampling algorithms presented in this work apply a heuristic search method to find a set of isoparametric curves which, when used to reconstruct the substitute geometry for the sampled surface [1, 2], produce a substitute geometry which features the lowest possible geometric deviation from the CAD model. This deviation measure (often characterized by the maximum distance between the substitute geometry, and the CAD model [1, 2]) is known as the sampling error, which is proportional to the measurement uncertainty [6].

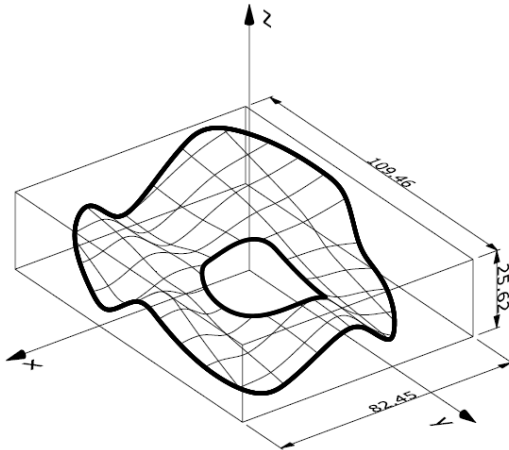


FIGURE 1. CAD model of a trimmed surface.

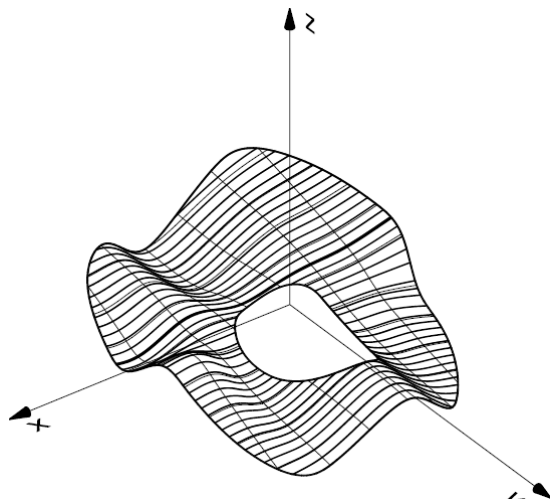


FIGURE 2. Sampling plan for model in figure 1.

The backbone for the sampling method presented in this work is the point-based representation of trimmed NURBS surface patches. In this representation, the trimmed surface patch is represented using a grid of points in the parametric domain. Each point in the point grid is represented using five parameters: 1) u , v parametric coordinates, 2) a logical operator (0/1) which indicates the occurrence of the point in the valid model space, and 3) the values of the surface curvature changes in the u , and v parametric directions, respectively. Using this representation, the surface isoparametric curves are ranked based on two measures:

1. The maximum change in surface curvature along the curve.
2. A measure of the curve occurrence within the valid model space.

The details of this representation and the ranking methodology may be reviewed in [2].

PROBE SCANNING PATH PLANNING

The topology of the CAD model for the physical object is used to determine the surface regions where the probe should be in contact with the part. These calculations are carried out in the parametric domain for the NURBS representation [4] of the surface. The results are then mapped to the model space. This is illustrated in figures 3, and 4, respectively.

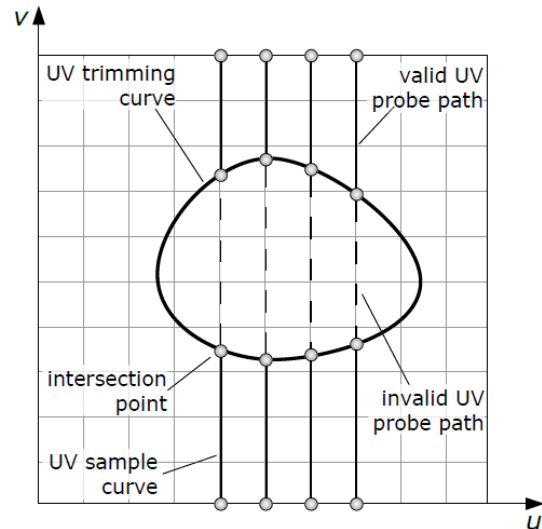


FIGURE 3. Probe scanning paths in parametric domain.

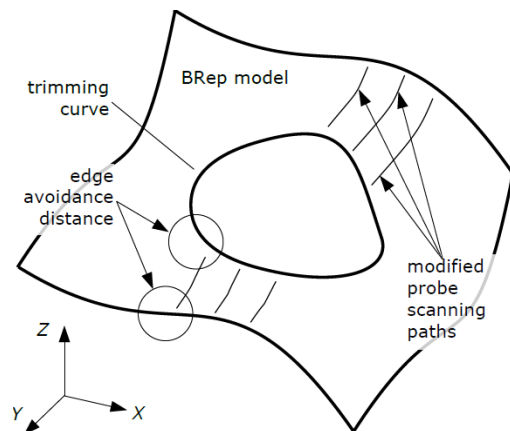


FIGURE 4. Modified probe scanning paths in the model space.

The probe paths are further cropped to provide an edge-avoidance distance for the scanning probe.

Following the above procedure, the probe scanning paths are sub-divided into sets of points, which are used to guide the probe during

the scanning process. The scanning of a free form surface using analog scanning is illustrated in figure 5.

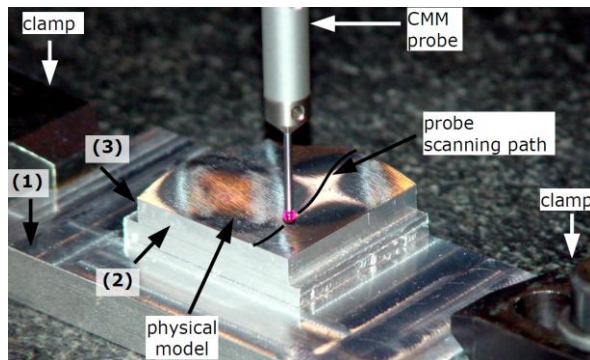


FIGURE 5. Analog scanning of a trimmed surface using the VAST probing system. Planes 1, 2, and 3 are used for part alignment.

DATA LOCALIZATION

Due to the many sources of error which influence the CMM measurement process, the measurement points, the measurement results always deviate from their CAD counterparts, often referred to as “nominal points”. This is illustrated in figure 6.

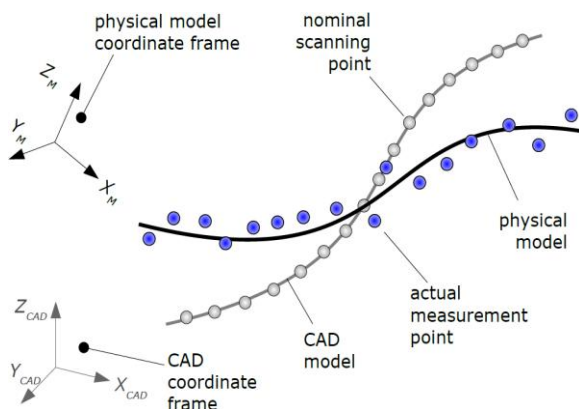


FIGURE 6. Nominal probe scanning path, and actual CMM measurement points.

In order to assess the physical model correctly, data localization is applied to the measurement points. Data localization is the process of calculating the optimum rigid body transformation such that the deviation function between the measurement data, and the CAD model is minimized. In this work, the Nelder-Mead simplex optimization algorithm is used, and the deviation function is the root-mean-squared error between the measurement points, and the CAD model.

The data localization procedure is carried out as follows:

1. Initial data localization: this step is applied in order to find a good initial solution for the optimization process. In this step, discrete non-collinear points are probed on the surface. The optimization process is applied to these points with respect to their nominal counterparts.
2. The Nelder-Mead simplex algorithm is applied to the measurement points with respect to the CAD model. The rigid body transformation obtained from the previous step is used as the initial solution to the optimization process.

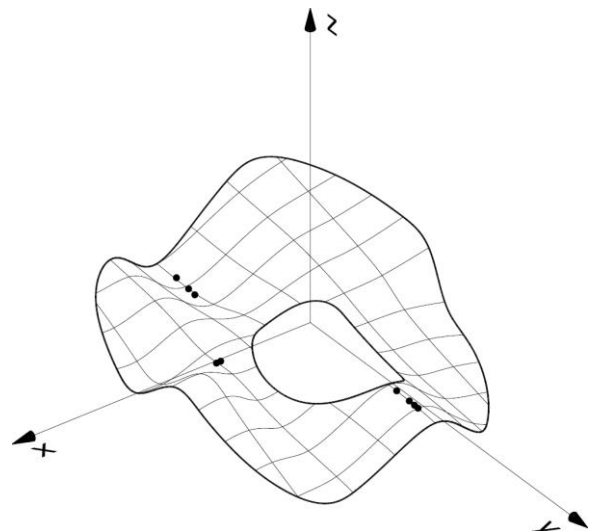


FIGURE 7. Data set used for initial localization.

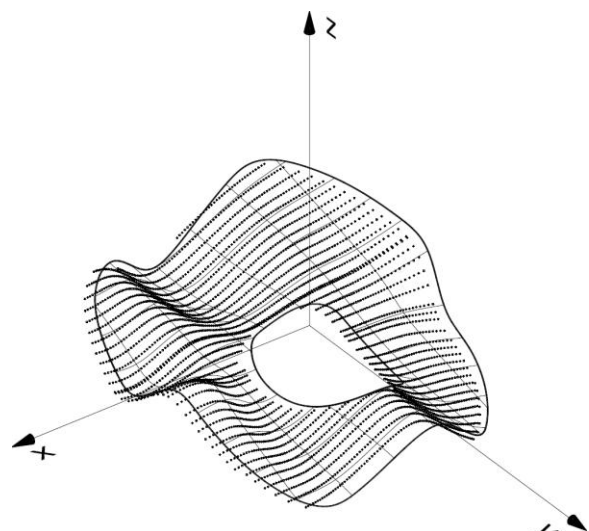


FIGURE 8. CMM measurement data.

The details of the localization procedure, including the criteria for the selecting points for initial data localization, and case studies may be reviewed in [7]. The data set used for initial localization, and the raw measurement points of a trimmed surface are illustrated in figures 7, and 8, respectively.

PROFILE ANALYSIS

The localized measurement points are used to construct the substitute geometry for the physical object [1, 2, 7].

The substitute geometry for the physical object is constructed as follows:

1. A NURBS curve is fitted [4] to each row of points which represent a single measuring path. A procedure is applied to account for missing curve regions [7].
2. A quadrilateral NURBS surface is fitted to the curve set obtained from step 1. The skinning algorithm [8] is applied in this work.
3. The trimming curves are extracted from the CAD model, and used to trim the surface obtained in step 2.

The resulting trimmed surface is used to represent the physical object in the profile tolerance analysis. An example is illustrated in figure 9.

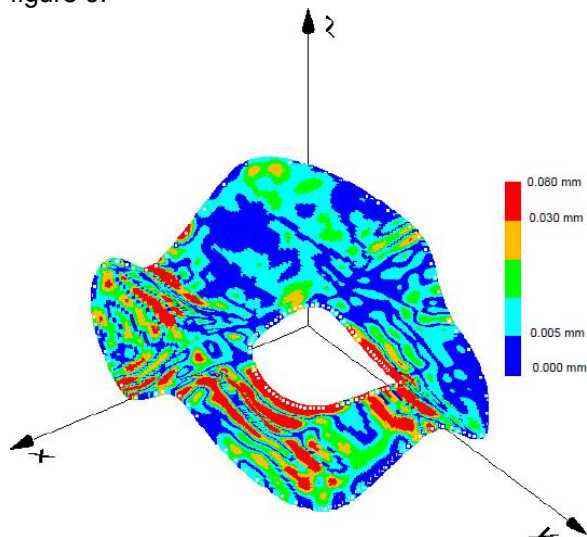


FIGURE 9. Profile analysis results.

CONCLUSION

A CAD-based, practical approach to CMM inspection is presented in this article. The novelty of this approach is in three aspects: 1) the use of the substitute geometry to represent the CAD model in profile analysis, 2) the

algorithms for sampling of free-form surfaces, and 3) the point-based approach for the representation of trimmed surface patches.

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A VIRTUAL MACHINE TOOL FOR THE EVALUATION OF STANDARDIZED 5-AXIS PERFORMANCE TESTS

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INTRODUCTION

Standards committees are developing standard test methods for the performance evaluation of 5-axis machining centers through the direct and indirect measurement of simultaneous motions of all five axes [1-2]. However, the compound effect of the geometric and servo control errors of all axes complicates the analysis of the measured data and the identification and separation of the individual error contributor(s). To better understand the effect of the individual errors on the measurement results, NIST is developing a multi-configuration five-axis Virtual Machine Tool (VMT) model.

BACKGROUND

Machine tool error modeling is not a new concept. Many models have been developed to investigate the effects of machine tools' geometric errors, stiffness, thermal behavior, and servo characteristics on workpiece geometry [3-6]. These models have also been used for the evaluation of measurement methods, for the evaluation of machine topologies, and for tool path compensation [5-8]. Research has shown that geometric errors are dominant at low speeds and servo errors are more dominant at higher speeds [9]. This paper presents the design of a generic virtual machine tool software platform that combines both geometric errors and servo errors for a variety of machine configurations.

VIRTUAL MACHINE TOOL

Developed using commercially available numerical computation software, the VMT is designed to be a generalized versatile model that can be used to simulate and analyze the combined or independent effect of the individual geometric and servo errors on standard 5-axis performance test results [1]. Simulations of machining tests [2,10] and multi-axis kinematic

tests [11] can be performed for many 5-axis machine tool configurations. In addition, the VMT may be used to predict the errors of workpieces machined by these types of machines.

The VMT is a software application that is the combination of six generalized computational modules (see Fig. 1). These are ideal trajectory generation (ITG), tool path animation (TPA), control error modeling (CEM), geometric error modeling (GEM), machine to workpiece coordinate system transformation (CST), and workpiece feature analysis (WFA). Versatility is achieved by populating the parameters of the generalized models with data described by machine specific information models [11] transferred using the eXtensible Markup Language (XML) [13].

Machine tool standards provide a common infrastructure that improves communication, efficiency, innovation, and interoperability through the standardization of terminology, measurement methods, analyses, and data formats [1,2,12,14-16]. The VMT is designed to be fully compatible with these standards by following their protocols, such as the identification and orientation of coordinate systems, the designation for the configuration of machine axes, the location of measurement functional points, and the formats for both machine and measurement data.

Machine tool information files are the core feature of the VMT, providing the ability to model multiple configurations and multiple machines. These files are generated following ASME B5.59-2, which defines electronic data formats for properties of machine tools including the machine tool performance data (e.g., linear positioning errors) [12]. Each file is machine

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specific and contains many data elements that describe the machine properties in a systematic structure. Examples of the types of elements defined in [12] include the machine's identification (ID), configuration, work zone, axis properties, performance, and errors. A short example of how data is stored is provided below as Example 1. This example demonstrates the format used for representing the mean unidirectional positional deviation of the X-axis for a positive approach direction.

EXAMPLE 1. X-axis linear positioning error, E_{xx} .

```
<MACHINE_ERRORS>
  <GEOMETRY>
    <AXIS_GEOMETRY>
      <AXIS_NAME>X</AXIS_NAME>
      <ERROR_TABLE>
        <MEASURAND>X</MEASURAND>
        <TARGETS>0.00 -10.00 -20.00 . . </TARGETS>
        <ERROR_DATA>
          <APPROACH_DIRECTION>POSITIVE</APPROACH_DIRECTION>
          <VALUES>0.00 -0.027 -0.020 . . . </VALUES>
        </ERROR_DATA>
      </ERROR_TABLE>
    </AXIS_GEOMETRY>
  </GEOMETRY>
</MACHINE_ERRORS>
```

The VMT is driven by simulation data files based on a general format similar to ASME B5.59 and written using XML. These files contain information used to identify the machine under evaluation, the workpiece, the workpiece location and orientation, the tool, the tool position relative to the workpiece (numerical control program), and simulation and analysis parameters. The parameters provide information pertaining to the analysis to be performed on the results, simulation speed, and Boolean type parameters used to activate and deactivate the various sources of machine error, e.g., servo control and geometric errors. The information pertaining to the machine, workpiece, and cutting tool are used as pointers for automatically acquiring data files and algorithms from various data repositories affiliated with the VMT. A short example of how errors are selected using "ON" and "OFF" statements is provided as Example 2. In Example 2, the linear positioning error for the X-axis, E_{xx} , is activated and the straightness error of the X-axis in the Y-direction, E_{yx} , is deactivated.

EXAMPLE 2. Error Selection Elements.

```
<ERROR_SELECTION>
  <GEOMETRY>
    <AXIS_ERRORS>
      <AXIS>
        <AXIS_NAME>X</AXIS_NAME>
        <ERROR_SELECTION>
          <MEASURAND>X</MEASURAND>
          <VALUE>ON</VALUE>
        </ERROR_SELECTION>
        <ERROR_SELECTION>
          <MEASURAND>Y</MEASURAND>
          <VALUE>OFF</VALUE>
        </ERROR_SELECTION>
      </AXIS>
    </AXIS_ERRORS>
  </GEOMETRY>
</ERROR_SELECTION>
```

The computational modules of the VMT are described below.

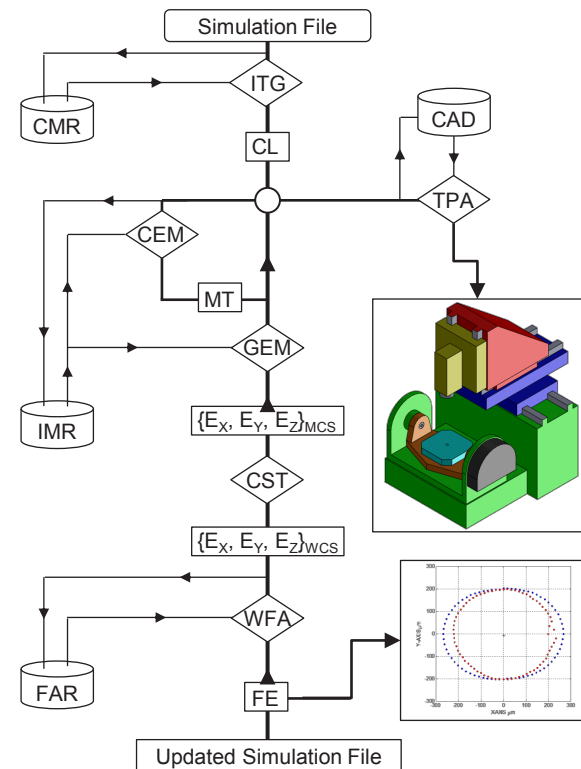


FIGURE 1. VMT Schematic

Ideal Trajectory Generation (ITG)

The numerical control (NC) program describes the cutting tool's nominal motion relative to the workpiece. The machine controller interpolates between the programmed points for each axis and moves the cutting tool and the workpiece along a smooth trajectory. Similarly, the VMT has a trajectory generation module with the

as the reference axis (primary axis). This information is also contained in the error data files. The geometric error module (GEM) uses such metadata to define the axes coordinate systems and to calculate the relative movements between each. To simplify geometric error modeling, the current version of the VMT supports only data files containing error data in which all measurements were made through one functional point. This method places all machine coordinate frames at the same location in the machine work volume. The effects of all angular errors are calculated from the same zero position to minimize the uncertainties associated with coordinate frame offsets. The output from the GEM is a tool path error file that contains the components (E_x , E_y , and E_z) of the error vectors between the tool and workpiece corresponding to the nominal positions along the tool path.

Coordinate System Transformation (CST)

The error data generated by the GEM is in machine coordinates (MCS). This data must be transformed to workpiece (instrument) coordinates (WCS) to analyze the effect of the machine errors on the workpiece geometry (measurement). This is accomplished in the VMT's coordinate system transformation module. Information describing the machine's configuration is used to transform the trajectory error data to workpiece (instrument) coordinates. The output from this module is a workpiece data file that contains the data used to describe the nominal tool path in workpiece coordinates and the workpiece errors, $\{E_x, E_y, \text{ and } E_z\}_{WCS}$, created by the machine tool.

Workpiece Feature Analysis (WFA)

Data from the workpiece error file is analyzed using algorithms from the feature analysis repository (FAR). These algorithms transform the workpiece error data to workpiece geometry or performance test parameters. The algorithms may include fitting circles, cylinders, and planes to the resulting data. The results from the analysis are then added to the simulation data file where it can be used by other software applications for future evaluation.

FINAL REMARKS

We provided a general description of a data driven virtual machine tool error model capable of simulating 5-axis performance test. Our VMT uses existing standards for test methods for machine tool performance as well as standards for representing machine data. Future work will

involve modeling and simulating multiple machine tool configurations, testing and evaluating 5-axis performance tests, and integrating the system with a commercially available computer aided manufacturing (CAM) software package. In addition, the model will be developed to support information models containing error data measured through multiple functional points within the machine work volume.

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SINGLE AND MULTIPLE-INSTRUMENT SURFACE CHARACTERIZATION WITH THE AREA STRUCTURE FUNCTION

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INTRODUCTION

For optical surfaces and transmitted wavefronts, it is often important to characterize errors in the spatial domain. Currently, power spectral density (PSD) is used to describe the spatial frequency characteristics [1,2], although there are potential pitfalls in its use. For example, the low spatial frequencies are removed before calculation, the calculation methods vary, and the results are sensitive to details like the windowing.

The linear structure function (SF) is less well known for characterizing surfaces, despite its first use 40 years ago for characterizing rough surfaces [3, 4]. Like the linear PSD, it loses information on anisotropy in the surface. Recent research suggests that the area SF has advantages over alternative characterizations of spatial frequency content [5]. In this paper we make the comparison between linear SF and area SF, apply the area SF to a diamond-turned surface, and combine data obtained with multiple instruments.

LINEAR / AREA STRUCTURE FUNCTION

The SF is the expectation of the squared height difference as a function of separation.

Figure 1 shows the calculation of the linear SF for area data [5]. Conceptually all possible point pairs of separation τ_1 are selected, their squared height differences calculated, and then averaged to obtain $SF(\tau_1)$. This process is repeated for all separations τ within the measurement area, and the whole SF is generated (Figure 1 (b)). For large numbers of points within the aperture, computational time increases rapidly but can be controlled by careful sampling. The SF can be computed over any chosen dynamic range for any aperture shape. Moreover, arbitrary separation into separate bands of figure, roughness and mid-spatial frequencies is unnecessary.

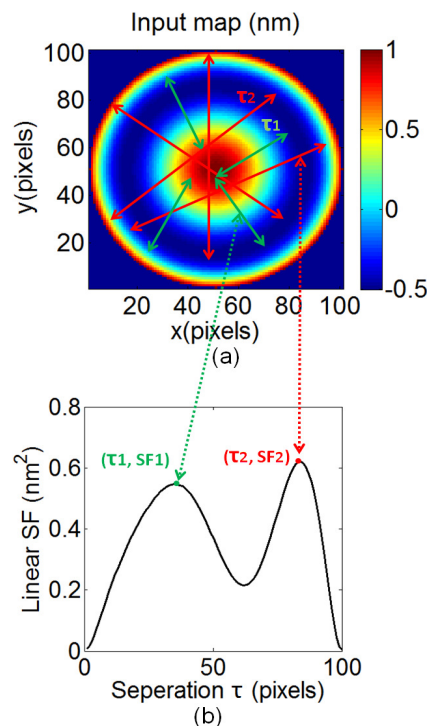


FIGURE 1. Calculation of the linear SF. (a) Input height map. (b) Linear SF as a function of separation.

The SF clearly shows the surface frequency content. If the surface has also been specified with a SF, then it is immediately clear what if any spatial frequencies need correction and hence what are the appropriate tools for the correction. The linear SF averages over all orientations for a given separation, effectively removing information on isotropy. The orientation information can be retained by calculating the area SF. Figure 2 (c) and (d) show the linear SFs for two surfaces generated with a single Zernike term; Figure 2 (e) and (f) show area SFs for the same surfaces. The two linear SFs are almost indistinguishable, while the area SFs clearly allow discrimination between the inputs.

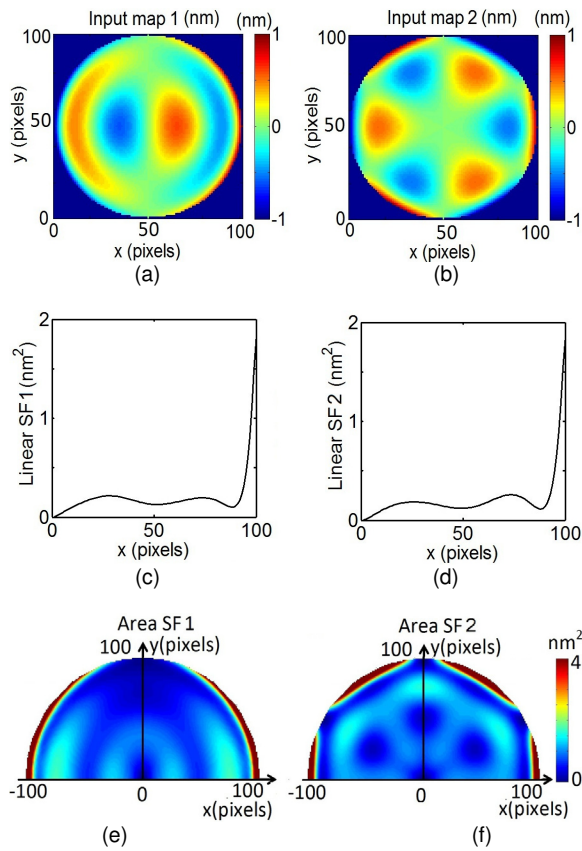


FIGURE 2. Comparison of linear and area SFs. (a) Input map 1. (b) Input map 2. (c) Linear SF of map 1. (d) Linear SF of map 2. (e) Area SF of map 1. (f) Area SF of map 2.

The area SF can be defined as [6]

$$S(\tau_x, \tau_y) = \frac{1}{(m-\tau_x)(n-\tau_y)} \sum_{i=1}^{m-\tau_x} \sum_{j=1}^{n-\tau_y} \{z(i, j) - z(i+\tau_x, j+\tau_y)\}^2 \quad (1)$$

where m and n are the number of equally spaced points in the x and y directions and $S(\tau_x, \tau_y)$ is the expectation of the squared height difference of points separated by τ_x, τ_y . For rotationally varying surfaces, this one-quadrant area SF is insufficient, because $S(\tau_x, \tau_y)$ is not equal to $S(-\tau_x, \tau_y)$. Therefore, a two-quadrant area SF was introduced.

SURFACE CHARACTERIZATION

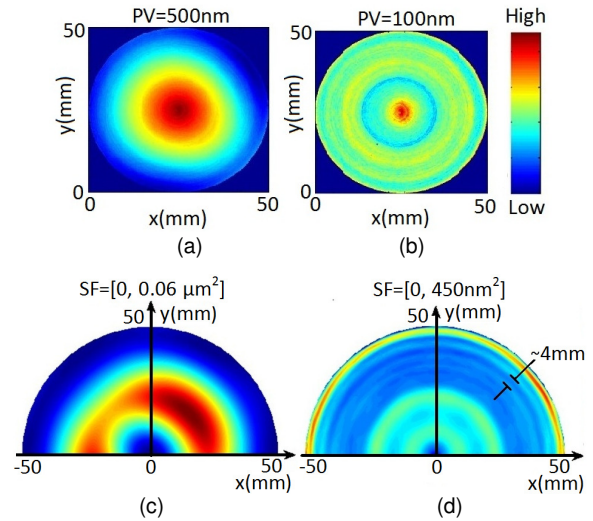


FIGURE 3. Diamond-turned aluminum. (a) Input map. (b) Residual error after removing the first 36 Zernike terms. (c) Area SF of the input map. (d) Area SF of the residual error.

Figure 3 shows both height maps and area SFs of a diamond turned aluminum flat measured with a Fizeau interferometer [5]. Figure 3 (a) shows the raw data and Figure 3 (c) the structure function for the data. Figure 3 (c) is dominated by 2 features: an orientation-independent peak at a separation of approximately 25 mm, and an asymmetry due to astigmatism in the part. The 25 mm peak is probably due to a squareness error in the machine giving a conical shape to the part. The astigmatism is most unlikely to be due to a machine error, but rather the result of mounting induced deformations.

Various methods may be used to aid in the visualization of mid-spatial frequency ripple. Here 36 Zernike terms are removed to obtain the residual error in Figure 3 (b), and the area SF of residual error is shown in Figure 3 (d). The annuli in the area SF at 10 mm and 22 mm separation are primarily due to the central high in the residual error (Zernikes do not fit a cone well) dropping to the low zone at a radius of 10 mm and then the edge that is again low. Note the highest frequency "ripple" in the area structure with a separation repeat length of about 4 mm. This ripple is rotationally invariant, that is the structure function values are constant for values of τ_x and τ_y for which $\tau = \sqrt{\tau_x^2 + \tau_y^2}$ is constant.

MULTIPLE-INSTRUMENT CHARACTERIZATION

Different instruments with varying lateral ranges and resolutions are sometimes used to characterize surfaces over a broad dynamic range. There may be advantages in combining data from multiple instruments in a single representation.

Representation With Log Scale

Figure 4 (a) showed a diamond-turned aluminum flat measured with a Fizeau interferometer (1000 pixels $\times$ 1000 pixels, Resolution=50 μm); and Figure 4 (b) shows the same flat measured with a Scanning White Light Interferometer (SWLI) (480 pixels $\times$ 480 pixels, Resolution = 0.22 μm). One way to represent the two measurements is shown in Figure 4 (c), which describes the spatial frequency content with separation from 0.22 μm to 50,000 μm . The horizontal and vertical axes are plotted on a log scale to emphasize the high frequency content. Diamond turning marks on the surface add high frequency content to the SF. Turning marks have radial symmetry and this appears as vertical frequency content in Figure 4 (c), given the global location of the SWLI measurement, as shown in Figure 4 (a).

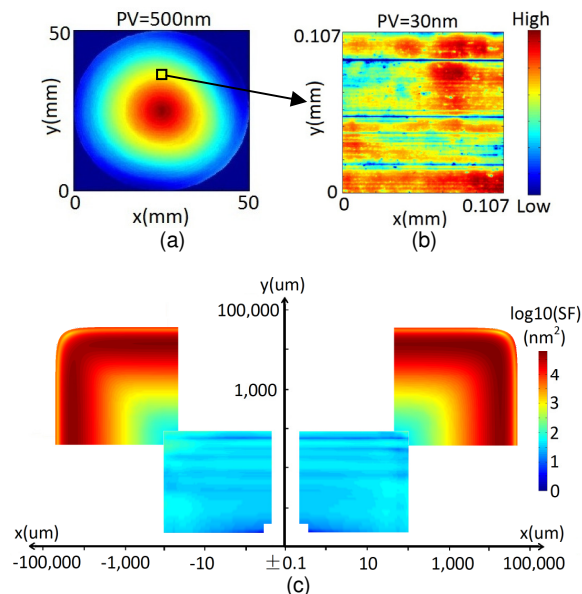


FIGURE 4. Diamond-turned aluminum. (a) Measurement with Fizeau interferometer. (b) Measurement with SWLI. (c) Combination of two area SFs.

Sampling Method

For position sensitive anisotropy, such as a diamond-turned surface, the SF depends on the measured position. However, if the SF is calculated for a collection of small-aperture measurements, the average is the correct SF for the high frequency content. Figure 5 (a) shows a simulated sinusoidal surface (2000 $\mu\text{m} \times 2000 \mu\text{m}$) with a radial wavelength of 50 μm ; and Figure 5 (b) and (c) show the area SF. Fifteen sub-apertures were picked randomly to simulate measurements taken with high magnification (200 $\mu\text{m} \times 200 \mu\text{m}$ areas). Then, the area SFs were calculated for each sub-aperture and then averaged (shown in Figure 5 (e)). The SF in Figure 5 (e) is an approximation of the exact SF in the high spatial frequency region shown in Figure 5 (c). Thus, averaging the area SFs from sub-aperture measurements using instruments with high spatial frequency resolution does represent the area SF in the high spatial frequency region.

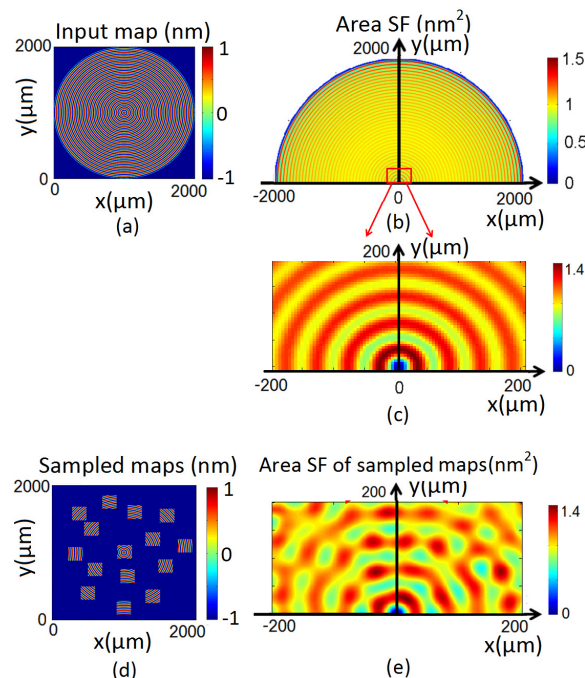


FIGURE 5. Sampling method. (a) Simulated sinusoidal surface. (b) Area SF of the simulated surface. (c) Zoom in on the area SF. (d) Randomly picked sub-apertures. (e) Average of the SFs of the sub-apertures.

Tilt Removal

In practice, however, the previous statement is not rigorously correct. This is because tilt in a

sub-aperture measurement is impacted by alignment uncertainty, and therefore removed from the measurement. Tilt leads to a quadratic contribution to the area SF. The correct area SF could be estimated by adding the right tilt to each sub-aperture measurement, perhaps estimated from the large-area measurement and knowledge of the global coordinates of each.

For example, Figure 6 (a) shows a simulated surface map that consists of power, coma and the same radial sine wave. Figure 6 (b) and (c) show its area SF. Figure 6 (d) shows the randomly picked sub-apertures, including the correct tilt for each, and Figure 6 (e) shows the averaged SF. We see that Figure 6 (e) is an estimate of Figure 6 (c).

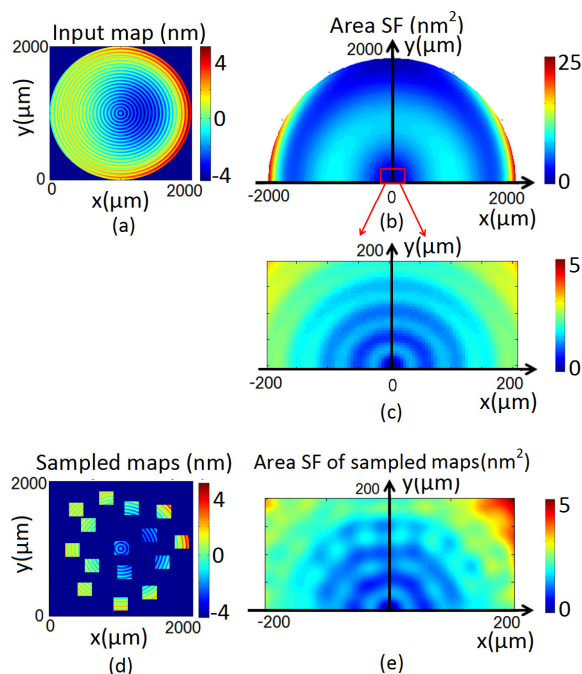


FIGURE 6. Tilt analysis of a simulated surface. (a) Simulated surface map with power, coma and a sine wave. (b) Area SF of the simulated surface. (c) Zoom in on the area SF in the high-spatial frequency region. (d) Randomly picked sub-apertures with tilt. (e) Average of the SFs of the sub-apertures.

However, if the tilt of the randomly picked sub-apertures is removed, the averaged SF would be approximately the same as that shown in Figure 5 (e). Notice that the wavelength of the sine wave is still identifiable, but the overall trend in the area SF is not correct.

CONCLUSIONS

The area SF contains anisotropic information, and it is useful for specifying and representing the spatial frequency content of a surface.

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Estimation of Volumetric Positioning Errors using Three-face Step-Diagonal Method

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INSTRUCTIONS

To ensure the motion accuracy of entire three-dimensional workspace such as machine tool or coordinate-measuring machine as many as 21 separate setups and measurements are needed for the linear, straightness, angular and perpendicular errors. In recent years, a method namely diagonal measurement following ISO 230-6 has been used for quick check on the volumetric errors of the machine. However, it is impossible to indicate the influence of each error among the linear, straightness and perpendicular errors of each axis from the results of diagonal tests; so, there is not enough information to make error compensation file to compensate volumetric errors.

THREE-FACE STEP-DIAGONAL METHOD

As FIGURE 1 shows, three-face step-diagonal tests in the XY-, YZ-, and XZ-planes and three linear motion errors in the X-, Y-, and Z-directions are used to estimate six straightness motion errors and three perpendicular errors (p indicates that the direction of motion is positive). Hence, 12 errors, including three linear errors, six straightness motion errors, and three perpendicular errors, can be estimated using the

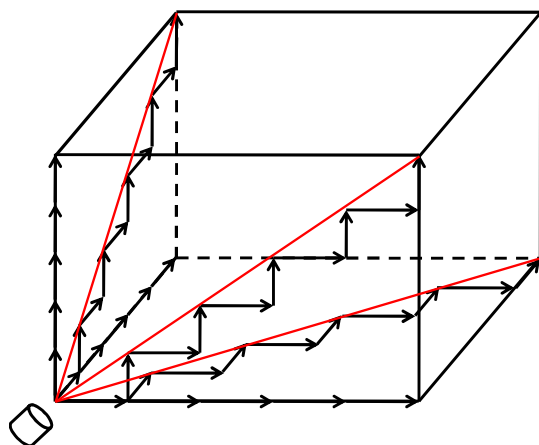


FIGURE 1. The six measurement paths

six data sets from six measurement paths shown in Figure 2. In practice, the laser source can be fixed at one corner of the workspace, while the flat mirror is moved to each of the six opposing corners, making the setup and alignment of the measurement process easier and faster. To provide data for the evaluation of the three-face step-diagonal method, experiments were conducted on a three-axis CNC machine with 3D workspace dimensions $X\ Y\ Z = 240 \times 240 \times 60$ mm. A direct measurement test for each error will be used to assess the accuracy of the estimates. This machine is based on motion control with a PID filter (PMAC2, Delta Tau) and a linear motor, and hydrostatic bearings are used to reduce the uncertainty of the machine, especially the angular errors. Therefore, it can be assumed

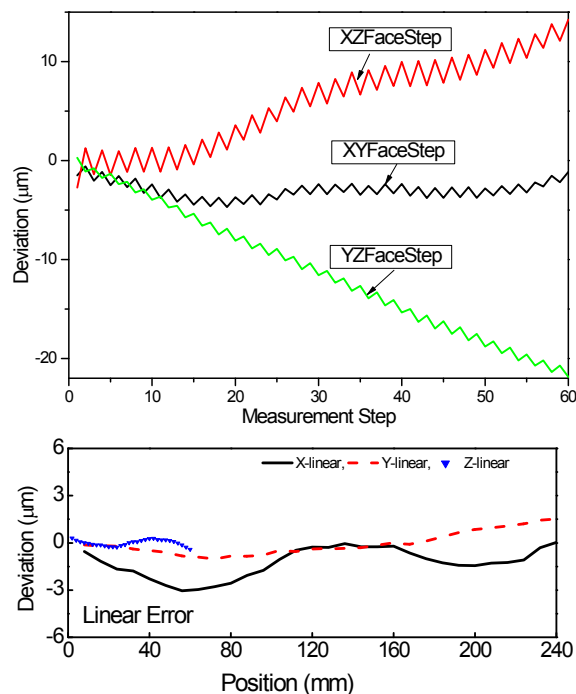


FIGURE 2. The three face step-diagonal and linear displacements

that the angular errors are negligibly small.

Estimation of perpendicular errors

Two-face diagonal measurements based on the ISO 230-6 standard are used to estimate the perpendicular error between two axes. Hwang et al. proved that the perpendicular error between the X- and Y-axes can be estimated not only from the length difference between the two face diagonals D_{pp} and D_{np} (based on the ISO 230-6 standard), but also from the face diagonal measurement D_{xy} and measured positions P_x and P_y on the X- and Y-axes, as shown in Figure 3a. Where D_{xy} , P_x and P_y are measured values including the ideal and error values. By using the processes employed in the XY-plane, the desired angles for the YZ- and XZ-planes can also be calculated from Eq. (1). In this equation, γ_{xy} , γ_{yz} , and γ_{xz} are the perpendicular errors between X-Y, Y-Z, and X-Z, respectively.

$$\begin{aligned}\gamma_{xy} &= \frac{D_{xy}^2 - P_x^2 - P_y^2}{2P_x P_y} \\ \gamma_{yz} &= \frac{D_{yz}^2 - P_y^2 - P_z^2}{2P_y P_z} \\ \gamma_{xz} &= \frac{D_{xz}^2 - P_x^2 - P_z^2}{2P_x P_z}\end{aligned}\quad (1)$$

By substituting the face diagonal and linear measured positions at the end positions into Eq. (1), the perpendicular errors between X-Y, Y-Z, and X-Z were found to be $-13.3 \mu\text{rad}$, $-422.8 \mu\text{rad}$, and $222.9 \mu\text{rad}$, respectively. On the other hand, using a master square and capacitive sensor, the directly measured perpendicular errors between X-Y, Y-Z, and X-Z were $-14.2 \mu\text{rad}$, $-420.7 \mu\text{rad}$, and $220.1 \mu\text{rad}$, respectively. Therefore, the maximum difference between the calculated and measured errors was $2.8 \mu\text{rad}$. Compared to direct measurement, the estimation procedure yields less accurate results, but is cheaper and faster in a practical environment.

Estimation of straightness motion errors

For the XY-plane, when the influence of perpendicular error ($2\gamma_{xy}P_xP_y$) can be eliminated from initial face step-diagonal data by using Eq. (1), the procedure is as follows:

The laser system will provide linear and straightness motion errors for the X-axis when the X-coordinate is moved, and for the Y-axis

when the Y-coordinate is moved. The combined motion errors for the X- and Y-axes can be obtained by repeating the measurements along the three-face step-diagonal of the plane. When the linear motion errors for the X- and Y-axes are directly measured, it is possible to estimate the contributions of the straightness motion errors for each axis, as well as the perpendicular errors between the X- and Y-axes. Figure 3b illustrates the three-face step-diagonal measurement setup for a single step, where e_{xx} and e_{yy} are the linear motion errors and e_{yx} and e_{xy} are the straightness motion errors of the X- and Y-axes, respectively. The first subscript of eab indicates the error direction and the second subscript denotes the axis of error. Δd_{xxy} and Δd_{yxy} are the respective diagonal displacement errors when X- and Y-coordinates are moved. S_x and S_y are the respective step lengths for the X- and Y-coordinates. The two angles in the figure can be calculated from Eq. (2).

When the X-coordinate is moved along the step-diagonal, the laser provides the combined data Δd_{xxy} , which includes the linear motion error e_{xx} and the straightness motion error e_{yx} . When a mirror alignment error δ_{s-xy} exists, the diagonal displacement error is given by Eq. (3).

$$\alpha_{xy} = \tan^{-1} \frac{S_x}{S_y}, \quad \alpha_{yx} = \tan^{-1} \frac{S_y}{S_x} \quad (2)$$

$$\Delta d_{xxy} = e_{yx} \sin \alpha_{yx} + e_{xx} \cos \alpha_{yx} + \delta_{s-xy} \quad (3)$$

From Eq. (3), the straightness motion error e_{yx} for the X-axis can be expressed by Eq. (4) for a single measurement step.

It is analogous to the formulation for X motion,

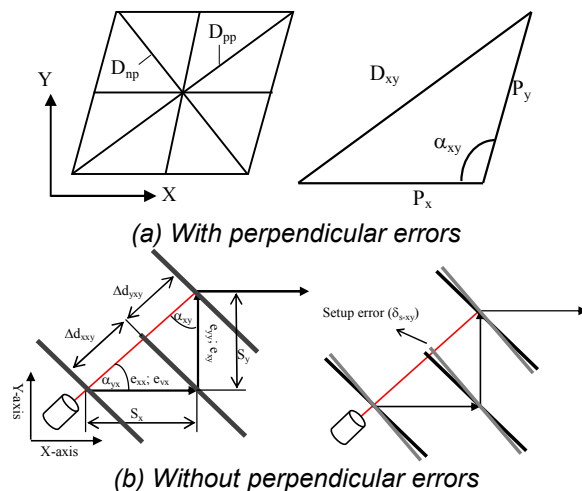


FIGURE 3. Errors of face step-diagonal

the straightness motion error e_{xy} for the Y-axis can be expressed by Eq. (5) for a single measurement step.

$$e_{yx} = \frac{\Delta d_{xxy} - e_{xx} \cos \alpha_{yx}}{\sin \alpha_{yx}} - \frac{\delta_{s-xy}}{\sin \alpha_{yx}} \quad (4)$$

The straightness motion errors are calculated using Eq. (4) and Eq. (5). All the parameters on the right-hand sides of these equations are already determined, except for the mirror misalignment δ_{s-xy} . This means that in every measurement step, the estimated straightness motion errors will include the mirror misalignment. Since the measurement step is small, δ_{s-xy} will not change greatly during the measurement process.

Hence, in n measurement steps, the accumulated misalignment error will equal $n \times \delta_{s-xy} / \sin \alpha_{yx}$ and $n \times \delta_{s-xy} / \sin \alpha_{xy}$ becomes the slope of the estimated straightness motion errors in Eq. (6) and Eq. (7). Fortunately, this slope can be eliminated by using a linear least-squares fit in the straightness calculation process, as has been noted in several papers [8–10].

$$e_{xy-cal}(n) = e_{xy-r}(n) + n \times \frac{\delta_{s-xy}}{\sin \alpha_{xy}}$$

$$e_{yx-cal}(n) = e_{yx-r}(n) - n \times \frac{\delta_{s-xy}}{\sin \alpha_{yx}}$$

Where $e_{yx-cal}(n)$ is the raw calculated straightness motion error for the X-axis including the mirror misalignment error, $e_{xy-r}(n)$ is the expected straightness motion error for the X-axis without the misalignment error, and $\delta_{s-xy} / \sin \alpha_{yx}$ and $\delta_{s-xy} / \sin \alpha_{xy}$ are the slopes from mirror misalignment. To obtain the straightness motion errors, the slopes from mirror misalignment in the initial calculated data are eliminated by using a linear least-squares fit, as in the example shown in Figure 4.

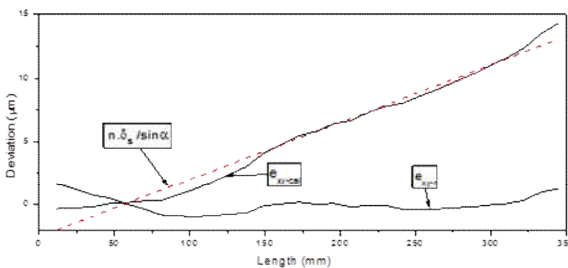


FIGURE 4. Example of a linear least-squares fit

The straightness motion errors in the YZ- and XZ-planes can be calculated similarly as XY-plane. Although flat mirror setup errors are included in the face step-diagonal data, they can be eliminated from the estimated errors by using a linear least-squares fit in the calculation process for the straightness errors. This means that the straightness errors can be estimated from the three-face step-diagonals and the three linear measurements, regardless of setup errors.

To evaluate the accuracy of the estimated errors, all straightness motion errors in the workspace were directly measured via a fiberoptic laser beam and optical master bar. The six directly measured straightness errors were compared with the calculated errors, and the results are shown in Figure 5. The maximum difference between the calculated and measured errors was $0.45 \mu\text{m}$. Hence, this estimation procedure will be useful for evaluating the straightness motion errors of a typical machine tool in an industrial application, comparing favorably with costly and time-consuming direct methods.

CONCLUSIONS

A three-face step-diagonal method for a 3D workspace was introduced, and its accuracy was demonstrated. Using this technique, a total of 15 errors can be calculated from three face diagonals and three linear displacements, including 12 estimated errors and three measured linear motion errors. The new procedure facilitates error measurement and reduces measurement time in comparison to Ibaraki's step-diagonal method [6], and also overcomes the limitations of the ISO 230-6 standard [1] and Wang's conventional step-diagonal method [2, 3].

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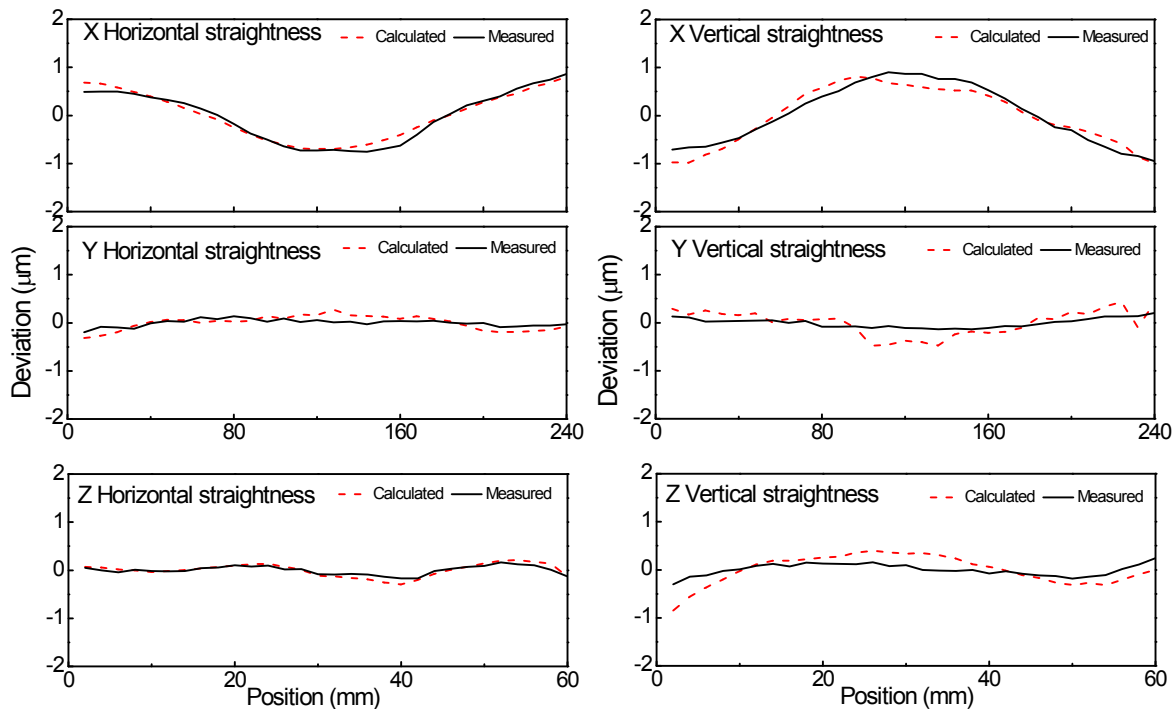


FIGURE 5. Comparison of calculated straightness errors with direct measurements

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MEASUREMENT AND CHARACTERIZATION OF 3D ASPHERICAL MICROLENS ARRAY FOR ADVANCED OPTICS

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INTRODUCTION

Micro lens array of aspherical surfaces has been adopted in a wide variety of optical applications such as backlight guide for LCD displays. The measurement and characterization of such micro lens array has attracted a lot attention from researchers and industries [1-4]. There are two methods for analyzing aspherical surfaces which include matching approach and data fitting approach, respectively [5]. If the CAD model of the micro lens array is known, matching approach is employed to align the measured data to the designed CAD model and hence the form error of the measured surface can be obtained by determining the deviations between the design surface and the measured surface.

On the other hand, if CAD model of the aspherical surface is unknown, data fitting approach is usually adopted for the characterization of the measured data. Most of the present characterization methods for aspherical surfaces are based on single profile measurement, which can not present the whole surface topography. Furthermore, there is relative little research has been reported on how to determine the optical axis and the vertex of aspherical surface, which are crucial for the post data fitting process.

As a result, this paper presents a systematic method for measurement and characterization of 3D aspherical micro lens array. A new method for the determination of optical axis and vertex of aspherical surface is also presented. Detailed fitting algorithms for spherical and aspherical surface fitting are explained. Experimental tests have been undertaken for an aspherical micro lens array and the measurement results are analyzed and discussed.

METHODOLOGY

Determination of Optical Axis and Optical Vertex

To measured aspherical surfaces, only tilting operation around X-axis (α) and Y-axis (β) is necessary due to its rotational symmetry.

To find the optical axis, the residual error (as indicated in FIGURE 1) is minimized as shown in Eq. (1)

$$FunRE(funT(D)) \rightarrow \min \quad (1)$$

Where, $funT$ is a rigid body transformation and is defined by

$$funT = Tr(\alpha, \beta) = \begin{bmatrix} \cos \beta & \sin \alpha \sin \beta & \cos \alpha \sin \beta \\ 0 & \cos \alpha & -\sin \alpha \\ -\sin \beta & \sin \alpha \cos \beta & \cos \alpha \cos \beta \end{bmatrix} \quad (2)$$

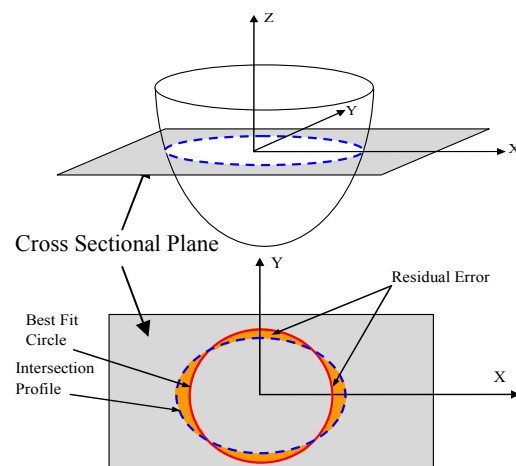


FIGURE 1. Graphical illustration for the process for the determination of the optical axis of the aspherical surface.

Afterward, the optical axis is the line vertical to cross sectional plane and across the center of the circle, the intersection between the optical axis and the surface is the vertex of the aspheric surface.

Fitting Algorithm for Aspherical Surface

FIGURE 2 shows the flowchart of the fitting process for an aspheric surface. The process starts with data fitting to find the best fit spherical surface, the center of which is used for the subsequent fitting of aspherical surface by centering operation. Afterward, aspherical parameters (e.g. base radius of curvature R and conic k) are firstly obtained by transforming nonlinear fitting to linear fitting algorithm.

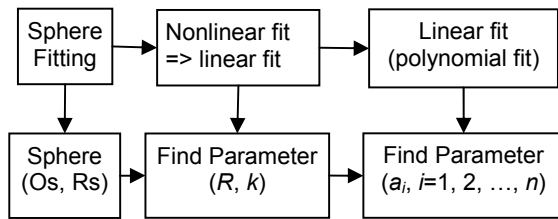


FIGURE 2. Flowchart for aspherical surface fitting.

The residual surface data between the aspherical data and aspherical surface data are used for determining the deformation parameters $(a_i, i=1, 2, \dots, n)$ by a linear fitting algorithm such as polynomial fitting. Hence, the aspherical surface parameters are obtained. The details of the algorithms involved are explained in the flowing sections.

Spherical Surface Fitting

Assume the spherical surface is expressed as

$$x^2 + y^2 + z^2 + Ax + By + Cz + D = 0 \quad (3)$$

Then it is a linear problem

$$Ax + By + Cz + D = -(x^2 + y^2 + z^2) \quad (4)$$

Or

$$\begin{bmatrix} x \\ y \\ z \\ 1 \end{bmatrix} \begin{bmatrix} A & B & C & D \end{bmatrix} = -(x^2 + y^2 + z^2) \quad (5)$$

Let

$$E = \begin{bmatrix} x \\ y \\ z \\ 1 \end{bmatrix}, X = \begin{bmatrix} A & B & C & D \end{bmatrix},$$

$$F = -(x^2 + y^2 + z^2) \quad (6)$$

Then Eq. (5) can be written as

$$EX = F \quad (7)$$

Therefore

$$X = E \setminus F \quad (8)$$

After parameters (A, B, C, D) are obtained, then the spherical surface is determined

$$\begin{cases} Os = (-A/2, -B/2, -C/2) \\ Rs = \sqrt{(A/2)^2 + (B/2)^2 + (C/2)^2 - D} \end{cases} \quad (9)$$

Aspherical Surface Fitting

Aspherical surface can be expressed as

$$z = \frac{cr^2}{1 + \sqrt{1 - (1+k)c^2r^2}} + \sum_{i=1}^n a_i r^{2i} \quad (10)$$

Where,

$$\begin{cases} c = 1/R \\ r = \sqrt{x^2 + y^2} \end{cases} \quad (11)$$

(i) Determination of c and k

Data fitting by equation

$$z = \frac{cr^2}{1 + \sqrt{1 - (1+k)c^2r^2}} \quad (12)$$

Eq.(12) can be rewritten as

$$r^2 = 2zR - z^2(1+k) = 2zR - z^2k - z^2 \quad (13)$$

Or

$$\begin{pmatrix} 2z & -z^2 \end{pmatrix} \begin{pmatrix} R \\ k \end{pmatrix} = r^2 + z^2 \quad (14)$$

Eq.(14) is a linear problem and can be solved easily.

(ii) Determination of $a_i (i = 1, 2, \dots, n)$

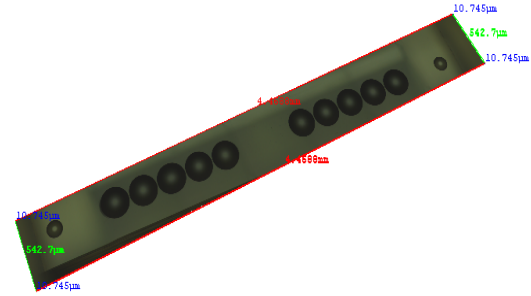
It can be solved by fitting

$$d_z = \sum_{i=1}^n a_i r^{2i} \quad (15)$$

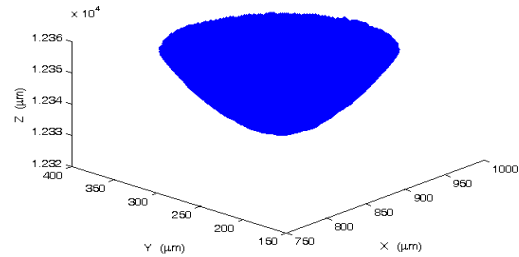
Where, d_z is the subtraction of the measured data by fitting of Eq. (12).

EXPERIMENTAL STUDIES AND RESULTS

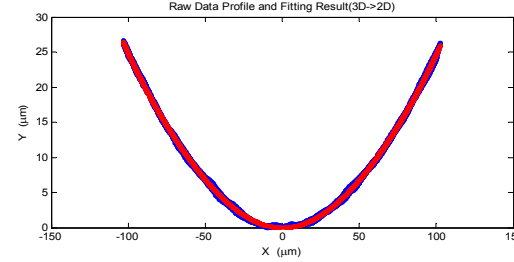
An aspherical surface is extracted from a lens array which was measured by Alicona IFM G4. The data was analyzed and the results were shown in FIGURE 3. FIGURE 3(a) shows the measured data for a microlens array while FIGURE 3(b) shows the data of one lens exported for analysis. The raw data of the profile and the results of the aspherical fitting results are shown in FIGURE 3(c); FIGURE 3(d) shows the residual errors after aspherical fitting of the raw data while the best fit aspherical surface based on the algorithms discussed in the previous sections is shown in FIGURE 3(e). FIGURE 3(f) depicts the error 3D topography of the microlens. As indicated in FIGURE.3 (f), the form error is found to be PV=1.7192 μm and RMS=0.23487 μm . Moreover, it also appears in a regular pattern in circumference direction, which might be due to the systematic error during the turning process. This is useful for the process error diagnosis and machining quality improvement.



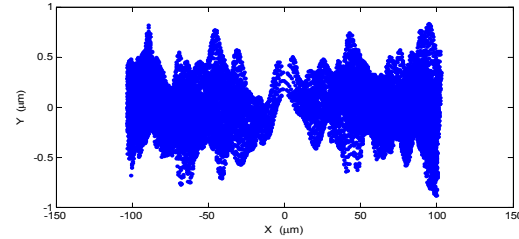
(a)



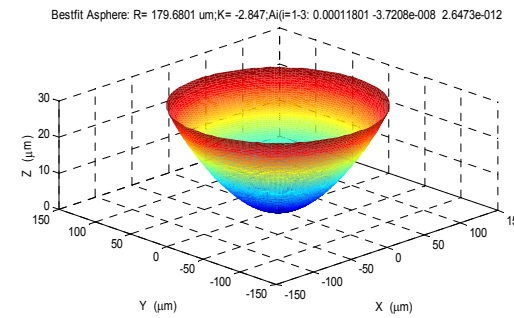
(b)



(c)



(d)



(e)

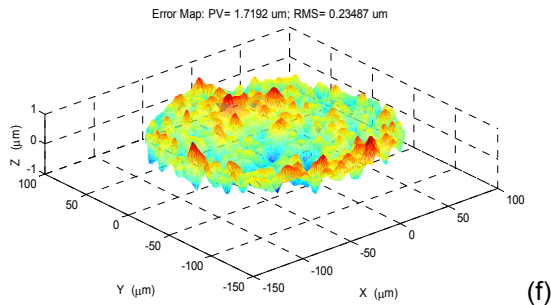


FIGURE 3. Form error of one aspherical surface extracted from aspherical microlens array: (a) Measured microlens array; (b) Lens data exported for analysis; (c) Raw data profile and aspherical fitting results; (d) Residual errors after raw data aspherical fitting; (e) Best fit aspherical surface; (f) Error 3D topography of microlens.

CONCLUSION

The use of microlens array of aspherical surfaces in optical applications has become more widespread and the measurement and characterization of microlens array has drawn a lot of attention by the researchers and industries. Most of the present characterization methods for aspherical surfaces are based on single profile measurement, there is relative little research reported on how to determine the optical axis and the vertex of aspherical surface, which are crucial for the post data fitting process.

This paper presents a systematic method for measurement and characterization of 3D aspherical microlens array, as well as a new method for the determination of optical axis and vertex of aspherical surface. Detailed fitting algorithms for spherical and aspherical surface fitting are also explained. Preliminary experiments have been undertaken for an aspherical microlens array and the measurement results are analyzed and discussed. The present method and characterization results are useful for error diagnosis for machining process of aspherical microlens array.

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Freeform Surface Analysis through use of Form Talysurf

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INTRODUCTION

General three dimensional surfaces on a part that do not have a rotational axis of symmetry, on or off the part, are known as freeform surfaces. In optics, freeform surfaces allow the optical designer many more degrees of freedom than traditional axi-symmetric surfaces and thus there is a growing interest in such surfaces in research and industry. While so-called slow and fast tool servo systems can generate some freeform surfaces, they are limited in slope and amplitude. Micromilling using newly marketed miniature diamond endmills allows more versatility and was the method used for generating the surfaces studied here. For this work, a 5-axis diamond machining center (Moore Nanotech 350 FG) was utilized to produce a unique freeform optical surface that was used in an Alvarez lens [6] and a test cubic surface, both in single crystal germanium.

Measurement of freeform surfaces is still an open research area. Most turning machines have an on-machine error measurement and correction procedure, but that is not the case for the multi-axis milling process used to fabricate the optics for this work. Through measuring the profile of a freeform surface using a form talysurf and given the analytical prescription for the surface, it is possible to manipulate the measured profile as a "rigid body" to determine the most likely location on the lens surface to within some uncertainty.

Freeform Surfaces

The first optic manufactured was a germanium cubic optic, described by Equation 1 [1].

$$z(x, y) = 0.4 [(x/10)^3 + (y/10)^3] \quad (1)$$

This cubic lens, Figure 1, was a preliminary test part to see if a more complex infrared optic, an Alvarez lens, was achievable through 5-axis diamond milling.

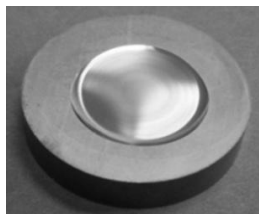


FIGURE 1. Ge cubic lens.

The overall design parameters for this lens were $\pm 100 \mu\text{m}$ surface variation, diameter of 10 mm, and an optical surface finish [1]. While this surface could theoretically be made using coordinated axis machining (slow and fast tool servo), the overarching goal of this work is to develop the process of freeform milling for use in even more complex surfaces than those reported here.

The second optic fabricated was an Alvarez lens, two plates which one surface being cubic and the other being planar make up the Alvarez lens system, described by Equation 2 [3].

$$z(x, y) = 0.0012 [(1/3)x^3 + xy^2] \quad (2)$$

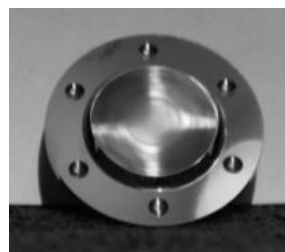


FIGURE 2. Alvarez lens cubic phase plate in Ge.

When the plates are inverted, relative to one another, and are superimposed in the z-axis, they create a variable focal length depended upon the different shifts between the two cubic phase plates [3].

Form Talysurf

Once each freeform optic was completed, the final surface form and finish were measured and compared to the theoretical surface requirements to determine if any error was caused during the machining process and to examine the effect of such errors on the optical function of the device. But measuring a freeform optical surface is difficult. Since the surface is machined to an optical finish, contact measurement methods, such as coordinate measuring machines (CMM's) and Form Talysurf machines, may damage the surface and further damage is convolved with the measurements since large deviations from a simple sphere or flat makes the use of a single null impossible. Further, while scanning white light interferometry (SWLI) can measure local finish, the need to remove higher order terms in

order to isolate finish introduces inherent and variable uncertainty. SWLI can also measure local form but is inherently limited by field of view and surface slope. For these reasons, a Taylor-Hobson Form Talysurf 120L was used to measure the optics. This system is capable of measuring a 120 mm x-y traverse length, 450 mm column (z) traverse length, with a stylus force less than 1 mN [2]. This contact method was chosen due to its low contact force, allowing the stylus to be used without significantly damaging the optic surface. A disadvantage of this system is that it does not currently have accurate positioning stages, the location of the measurements is only generally known.

Thus, the data was analyzed using MATLAB to determine each surface's form, profile, waviness, and roughness. Earlier work determined if any machining errors were present in the optic's surface through filtering the data, using a discrete Fourier transform, and then comparing it to the theoretical surface. To help eliminate uncertainty with part orientation, the next step was to take data for a trace across the cubic lens surface and use MATLAB to match it back to a most likely surface location by minimizing the errors relative to the expected theoretical form.

MEASUREMENT PROCEDURE

To determine the optic's form, finish, and waviness, trace measurements were taken using the following parameters: 12 mm data length, 0.3 mm run-up length, and a measurement speed of 5 mm/s. In order to avoid any un-anticipated damage to the Alvarez lens surfaces, all work thus far has used only the germanium cubic test surface.

Five traces were taken across the cubic test surface. Each of the five traces was taken with approximately 45 degrees clockwise rotation between each with the fifth measure roughly repeating the first, as shown in Figure 3. Rotations were executed by hand, and therefore the exact positions are not known.

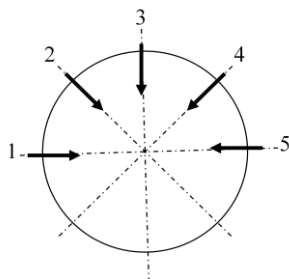


FIGURE 3. Cubic test surface form talysurf measurement locations.

SURFACE FINISH ANALYSIS

In order to analyze the surface roughness, waviness, and form, a discrete fourier transform filter was applied, through the use of MATLAB. This separated the real data into waviness and roughness profiles, with the form removed as seen in Figure 4. Using the waviness and roughness profiles, the RMS waviness and RMS roughness values were determined and compared to previous SWLI measurements.

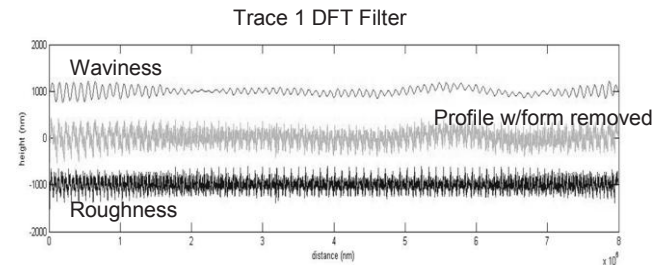


FIGURE 4. Cubic test surface DFT results for 1st talysurf trace - waviness is top line, roughness is bottom line, and profile with form removed is middle line.

FORM ANALYSIS

The actual form errors in the data have not yet been determined. Instead, a simplified algorithm has been developed and will be used on the actual data in future work. The results of the simplified algorithm are presented here. As a starting point, two-dimensional form analysis of the surface's profile has been undertaken. The algorithm for this analysis is as follows. First, a two-dimensional profile of the theoretical surface was obtained by setting $y = 0$ in Equation 1. A simulated test data-set was then generated from the 2-D profile and rotated and translated in two dimensions to simulate the effect of the unknown measurements locations in the real talysurf data, as shown in Figure 5. The algorithm then implements an optimization scheme that used a Homogeneous Transform Matrix (HTM) to translate and rotate the simulated data until the RMS errors are minimized relative to the theoretical profile.

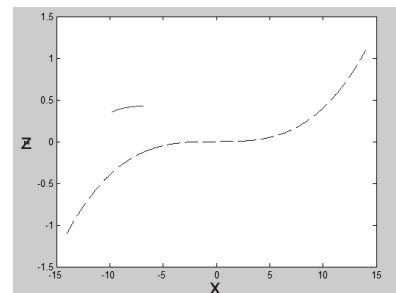


FIGURE 5. Cubic surface profile and slice segment generated using MATLAB.

To locate the slice onto the original cubic profile, three-variable minimization was implemented, using the x-coordinate, z-coordinate, and rotational angle as the minimization variables, minimizing the RMS error between the x-, z-, and angle coordinates. Multiple methods of minimization have been considered, however, the results from a Hooke-Jeeves minimization routine are presented here. A Hooke-Jeeves method was chosen because it was a direct search technique “based on a sequence of exploratory and pattern moves [5]”. The method is generalized to more dimensions (potentially up to six in the analysis of the real data) and it does not require data on the local slope of the minimization function. Further, since this is a precision-metrology application it is necessary to be able to control the minimization so that the final results are in the range of the expected surface errors (less than 1 micrometer at each point). As can be seen in the results, this requirement is challenging and requires more work. For details of the Hooke-Jeeves method, see reference [5].

RESULTS

Surface Finish Analysis Results

From the 5 traces shown in Figure 3, the RMS roughness and waviness results are listed below in Table 1. From these results, and overall mean RMS roughness of $113 \text{ nm} \pm 13 \text{ nm}$ and a mean RMS waviness of $64 \text{ nm} \pm 26 \text{ nm}$ were determined. The RMS roughness of the SWLI ($181 \text{ nm} \pm 36 \text{ nm}$ [4]), the SWLI result was slightly greater. This difference is most likely introduced by uncertainty in the fitting algorithm used to remove the higher order terms and to isolate an RMS value.

TABLE 1. Ge cubic test surface Form Talysurf RMS roughness and RMS waviness results.

Trace #	RMS Roughness (nm)	RMS Waviness (nm)
1	117	81
2	133	21
3	112	60
4	96	89
5	109	68

The filtered surface profile and waviness curves show surface imperfections and possible tool wear pattern. Material buildup on the tool was seen as patterns of decreased amplitude on the waviness curves. Surface scratches and a tool

centering error appeared as randomly interspersed amplitude spikes on the filtered profile curves.

Form Analysis Results

The results of minimization are shown in Figure 6 and represent an RMS error of 0.1946 mm using the parameters listed below in Table 2. The best fit slice converges the same orientation as the theoretical profile, but does not directly overlay with the theoretical profile, x and z errors remain. The dynamics of the minimization routine are summarized in Figure 7.

TABLE 2. Parameters used to achieve RMS error of 0.1946 mm.

Parameter	Value
$[x_o, z_o, \theta_o]$ (initial guesses)	[0,0,0]
x_b (Base Point)	[0,0,0]
$[h_{x0}, h_{z0}, h_{\theta0}]$ (step size)	[0.1,0.1,0.5]
eps (tolerance)	0.001

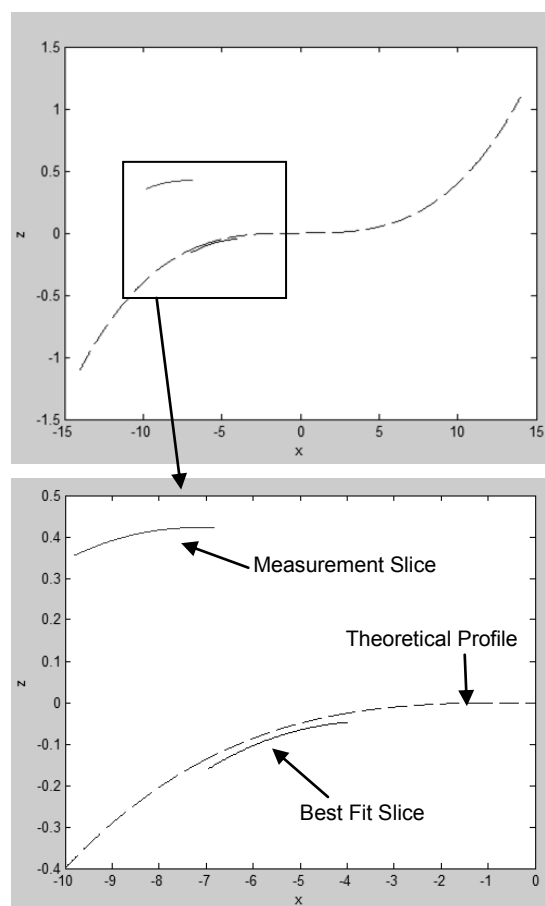


FIGURE 6. Measurement slice best fit location on the theoretical cubic profile.

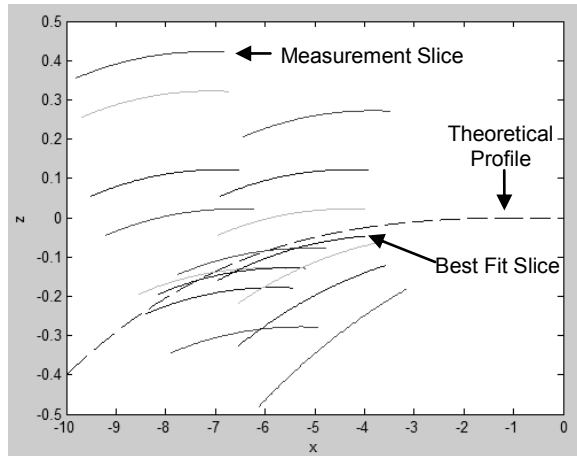


FIGURE 7. Representation of the steps taken by the algorithm to achieve the final slice fit. The initial measurement data in the upper left corner and the final slice fit is the slice closest to the theoretical profile (dashed line).

CONCLUSIONS AND FUTURE WORK

The RMS roughness and RMS waviness determined using data gathered from the Form Talysurf was comparable to that of the SWLI, but can yield more information, such as machining imperfections in the part surface and tool wear, through further analysis of the traced profile. Overall, this method showed that form could be determined through fitting a profile to the measured data and comparing to the theoretical surface, but too many uncertainties were present due to unknown orientation of the part during data collection.

To aid in eliminating uncertainty due to part location during measurement, it was seen that the profile of a freeform, complex surface can be measured and fit back to its approximate location onto the theoretical profile with minimal error, within a set tolerance. From the analysis of the germanium cubic lens, the proposed algorithm was able to achieve a best fit, but using this minimization routine, the results strongly depends on the value of the step size. This can be seen in Figure 5 where the best fit slice does not directly overlay on the theoretical profile. The gap between the two profiles could be greatly reduced if the step size could be optimized.

Future work would include determining a method for optimizing the value of the step size for each parameter to achieve error minimization without arbitrarily assigning those values. Also, this minimization routine can be adjusted to do surface error minimization (3-dimensional) for the types of optics used for this work. But overall, the algorithm was successful locating the measurement slice onto the theoretical profile by minimizing the RMS error.

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Uncertainty of the Interferometric Radius Measurement in a Production Environment

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INTRODUCTION

The uncertainty evaluation of the interferometric radius measurement is well documented in literature [1-5]. Schmitz, et. al. [2], gave a partially through review of the evaluation the interferometric radius measurement uncertainty of a nominally 24.466 mm diameter sphere at NIST. Other work [3-5] showed full or partial uncertainty evaluations for measurements done at UNC Charlotte. In many cases, full uncertainty examples which follow the GUM (Guide to the Expression of Uncertainty in Measurement [6]) shown in literature are from single time measurements at national laboratories or academic environments. Applying the same principles to a production manufacturing environment will have complications and is not typically demonstrated in literature.

This paper will show how the uncertainty evaluation must be considered for a production manufacturing environment. This will focus on the changing conditions that require a new evaluation of the uncertainty. In addition, because measurements in a production environment are taken by many different people (who may have different skill levels), we must also have an easy to use tool that the operator can quickly and easily calculate the uncertainty of any specific measurement.

INTERFEROMETRIC RADIUS MEASUREMENT

Figure 1 shows a schematic of the interferometric radius measurement on a Fizeau type interferometer. The interferometer uses a transmission sphere (in black) to produce a converging spherical wavefront. The part, here concave, is first placed at confocal, where the test wavefront and part curvature match. The location of this position is recorded. Then, the part is moved to cat's eye, at the test wavefront focus and this position is recorded. The difference between the two positions is the radius. The procedure is the same for a convex part; the confocal position is in the converging wavefront close to the interferometer.

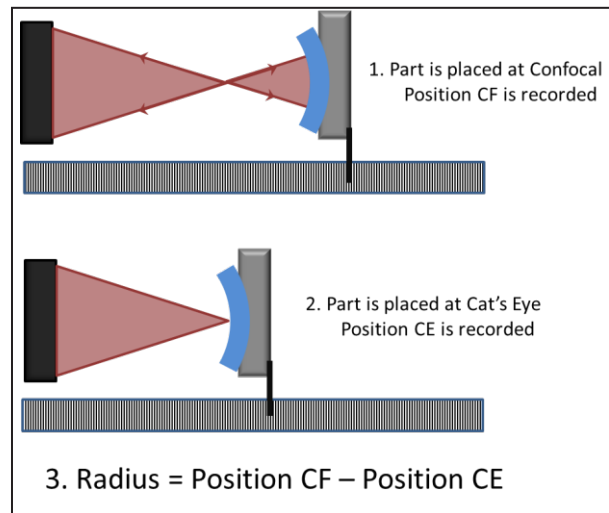


FIGURE 1. Schematic of the interferometric radius measurement, a two part measurement.

The locations can be measured using various methods, which will contribute to the final uncertainty. In the production environment, DMI (displacement measuring interferometers are used).

THE PRODUCTION ENVIRONMENT

In the production environment, the following conditions can vary part to part, which sometimes means hour to hour.

- Part Radius
- Part Diameter
- Holding Conditions
- Transmission Sphere f/#
- Transmission Sphere Diameter
- Transmission Sphere Model Number
- Interferometer Model Number
 - Source Type
 - Camera Resolution
- DMI Model
- Environment
 - Temperature/Pressure/Humidity
- Measuring Procedure (Step through null or software correction)
- Operator, Shift

In addition to the aspects listed above, there are conditions that may change over time, such as error motions and alignments.

UNCERTAINTY CONTRIBUTORS

The uncertainty contributors that are considered are:

- Stage error motions
- Cosine error
- Abbe error
- Unsensed length due to temperature variations
- Deadpath error
- Uncertainty in environmental sensors
- Turbulence
- DMI system
- Fizeau transmission sphere error
- Surface figure error
- Viewing aperture
- Locating the null position
- Model used

Additional Considerations for the Manufacturing Environment

For some uncertainty contributors, we can regularly record values that can then later be used in uncertainty evaluations. These include measuring the stage error motions, the DMI cosine error, and the temperature / pressure / humidity sensor uncertainties. These are measured and recorded during regularity scheduled interferometer maintenance.

For the production environment, we must develop an easy to use tool for operators to evaluate the uncertainty on a case by case basis. This may be something as simple as an Excel file which does the calculations, or a specialty built software program which collects data from the interferometer maintenance and also calculates the uncertainty.

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EVALUATION METHOD OF ACCELERATION CORRELATED POSITION ERRORS IN MACHINE TOOLS

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INTRODUCTION

Machine tools have to reach very high levels in accuracy and speed. The detailed analysis and systematization of machine tool dynamics with respect to these properties is ongoing research topic as these properties are prerequisites to achieve dynamic path-accuracy. For the distinction of the sources causing dynamic, especially acceleration dependent, errors several works ([1], [2] and [3]) have been carried out, which are stating a systematic and repeatable behaviour of dynamic displacements in models and measurements. These conclusions suggest the modelling and the compensation of errors due to machine dynamics. Probable solutions are on-line simulation of machine [4] and state control [5]. These methods permit the complete control of dynamic components, including p.e. damping and friction, but have due to parameter-uncertainty high requirements on approaching feasible settings to ensure robustness.

Based on the stated systematics of dynamic displacements, especially on acceleration level, this paper presents a method to describe these systematics. With this method, the measured displacements can directly be correlated to current acceleration. To avoid the influence of errors on position and velocity levels and errors due to excitation (jerk), a suitable measurement motion is proposed.

MEASUREMENT METHOD

The evaluation procedure consists of measuring circular movements of the tool center point (TCP) and comparing these movements with internal measurement values. Due to this combination the cross-talk and the in-talk parameters, which are elucidated in the following sections, of a machine tool can be derived. The measurement motions are circular movements the three main planes in the machine coordinate system, carried out by the interpolated movements of the linear axes of the machine tool.

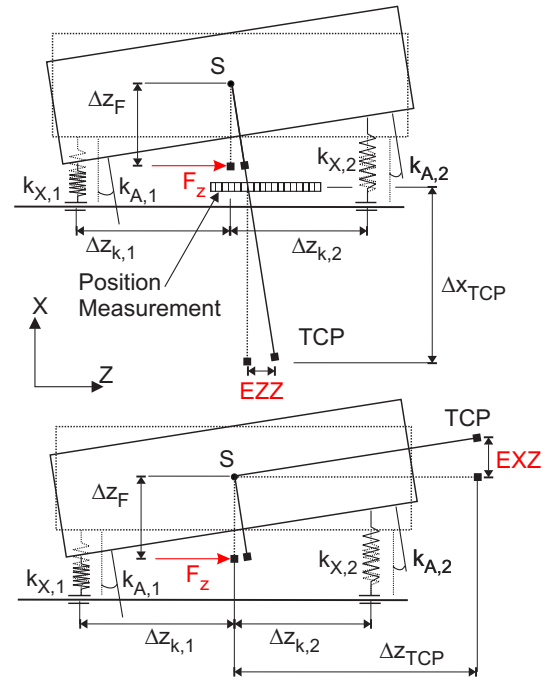


FIGURE 1. Schematic of the inertial tilt effects on the TCP displacements. The drive force F_z causes the in-talk EZZ (above) and cross-talk EXZ (beyond).

Measurement motion

Especially the two states, acceleration and jerk, have to be considered in detail. The acceleration and the offset between the center of gravity and drive force input position causes a momentum on the acting machine tool structure [6]. The momentum leads to tilt motions, as inertial in-talk and cross-talk (figure 1), which are defined in equations (1) and (2):

$$EZZ = \frac{F_z \Delta z_F \Delta x_{TCP}}{\sum_i k_{X,i} \Delta z_{k,i}^2 + \sum_j k_{A,j}} \quad (1)$$

$$EXZ = \frac{F_z \Delta z_F \Delta z_{TCP}}{\sum_i k_{X,i} \Delta z_{k,i}^2 + \sum_j k_{A,j}} \quad (2)$$

Assuming linear stiffness, the magnitude of the cross-talk can be linearly correlated to the acceleration. The jerk defines the time span for the development of the reachable acceleration [3], which can easily be seen by the approximation of the drive force $F_z \approx m \cdot \ddot{z}$ and its insertion in the equations (1) and (2). The time span of this dynamic input leads to the excitation of the machine. To distinguish the cause of dynamic effects due to jerk and acceleration, circular motions provides low frequencies of excitation. As an example, the circular motion with radius of 2mm with an acceleration 5m/s^2 yields 9Hz for the excitation of an axis.

External measurements

The external measurement of the circular motions denote relative displacements between TCP and work piece point (WPP). As measurement device, the R-Test is used providing simultaneous spatial displacement values combined with moderate programming and set-up effort required.

R-Test device

The R-Test [7] is a measurement set-up, where the TCP is represented by a precision sphere and the WPP by three orthogonal incremental probes with flat probing tips, see figure 2. The measurement uncertainty of the R-Test measurement setup is $U(k=2) = 2.8\mu\text{m}$ for a measurement range of 2mm . After calibration on the machine tool the probe values can directly be transformed in machine coordinate system. The measurement range is $\pm 3\text{mm}$.



FIGURE 2. The R-Test set-up

Internal measurements

In addition to the TCP-displacements obtained from the R-Test measurements the corresponding values at the internal measurement systems

are used. The use of close-loop position measurement is a prerequisite for the approach outlined here. The difference between the values obtained at the direct measurement systems of the axes and the external measurement describe the structural error, which can not be seen by the control system of the machine and can not be compensated or influenced by the control loop of the machine tool.

EVALUATION METHOD

The evaluation of measured circular motions exemplary in the ZX-Plane is explained next. The set points (z_{set}, x_{set}) and the set acceleration $(\ddot{z}_{set}, \ddot{x}_{set})$ of circular motion are given by:

$$z_{set} = r \cos(\omega t) \Rightarrow \ddot{z}_{set} = -r\omega^2 \cos(\omega t) \quad (3)$$

$$x_{set} = r \sin(\omega t) \Rightarrow \ddot{x}_{set} = -r\omega^2 \sin(\omega t) \quad (4)$$

with nominal radius r and angular velocity ω . The position and the acceleration are in phase. The internal and external measured circular motions can get normalized with the measured coordinates and multiplied with nominal radius r to get the nominal circle as stated in equations (3) and (4). So, the allocation between set points and measured values is ensured. Furthermore the actual accelerations at the measured positions, based on the internal measurement, are obtained. Assuming small displacements, which is given with required accuracy in micrometers of stiff machine tools with large axis strokes, the actual points (z_{actual}, x_{actual}) are stated as:

$$z_{actual} = z_{set} + EZ\ddot{Z}\ddot{z}_{set} + EZ\ddot{X}\ddot{x}_{set} \quad (5)$$

$$x_{actual} = x_{set} + EX\ddot{X}\ddot{x}_{set} + EX\ddot{Z}\ddot{z}_{set} \quad (6)$$

with position errors $EZ\ddot{Z}, EX\ddot{X}$ in axis-direction (in-talk) and $EX\ddot{Z}, EZ\ddot{X}$ orthogonal to axis-direction (cross-talk) presented in following sections, are assumed to be linearly to the accelerations. The effects of cross- and in-talk have to be separated. Firstly the cross-talk is analysed. this leads to corrected values without cross-talk influences which are then further processed to obtain the in-talk parameters.

Cross-talk

The cross-talk values $EZ\ddot{X}, EX\ddot{Z}$ are calculated out of the planar inclination of the two other circular motions in XY and YZ-plane using "least square plane fitting". With the obtained planar fitting parameters c and d , the acceleration correlated inertial cross-talk errors yields:

$$EZ\ddot{X} = \frac{c}{\ddot{x}_{max}} \quad (7)$$

$$EX\ddot{Z} = \frac{d}{\ddot{z}_{max}} \quad (8)$$

Figure 3 illustrates the effect of $EX\ddot{Z}$.

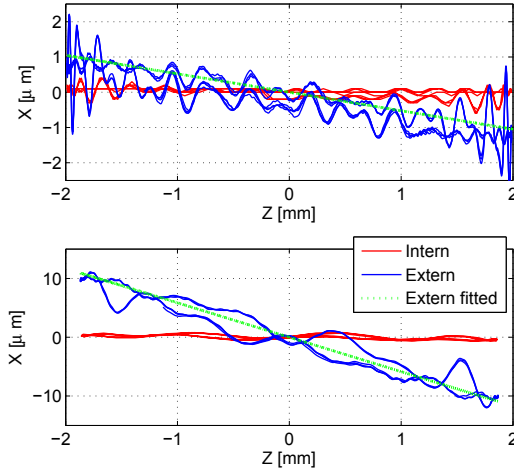


FIGURE 3. View on ZX-Plane of internal and external measurement of circular YZ-Motion, above: $a_{max} = 0.5m/s^2$, below: $a_{max} = 4.5m/s^2$.

In-talk

The in-talk values $EZ\ddot{Z}$ and $EX\ddot{X}$ obtained with "least square ellipse fit". The cross-talk values are evaluated in the previous section, hence the equations (5) and (6) are modified to:

$$z_{actual}' = z_{actual} - EZ\ddot{X}\ddot{x}_{set} \quad (9)$$

$$x_{actual}' = x_{actual} - EX\ddot{Z}\ddot{z}_{set} \quad (10)$$

and evaluated with "even ellipse fitting".

The resulting semi-minor axis value b and semi-major axes value a are given in machine axis direction. The in-talk values are obtained by

$$EZ\ddot{Z} = \frac{r - a}{\ddot{z}_{max}} \quad (11)$$

$$EX\ddot{X} = \frac{r - b}{\ddot{x}_{max}} \quad (12)$$

with the nominal radius r and the maximal reached accelerations $\ddot{z}_{max}$ and $\ddot{x}_{max}$.

RESULTS

Figure 4 shows the fitting of the measurements. The assumption of (5) and (6) is visible and distinct for higher accelerations, where the dynamic displacements are higher in amplitude. Displacements due inertial cross-talk and in-talk are proportional to acceleration. In the next section, evaluation results over different parameter settings are presented and discussed for illustration.

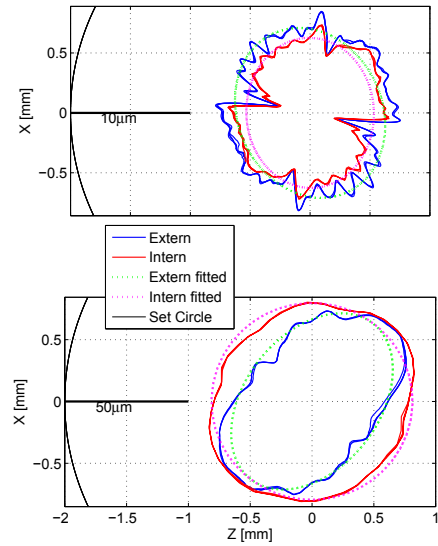


FIGURE 4. Magnified external and internal measurements and fitted ellipses in reference to the set circle ($r = 2mm$), above: $a_{max} = 0.5m/s^2$, below: $a_{max} = 4.5m/s^2$.

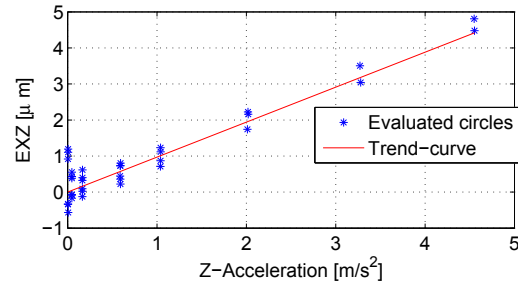


FIGURE 5. Evaluated cross-talk values with trend curve for different parameter settings over acceleration.

To discuss the influence of different parameter settings, the circular motions have been evaluated with different jerk, acceleration and velocity. The clockwise and counter-clockwise direction of the circles in the three main planes leads to further parameters. Figure 5 shows exemplary cross-talk values dependent on the maximally reached accelerations of the measurements. The evaluated values show mainly acceleration dependency. Static and geometric effects like component errors influence the evaluation of the parameters for low acceleration. The resulting proportionality parameters show higher dispersion, as seen in figure 6. Tables 1 and 2 show the re-

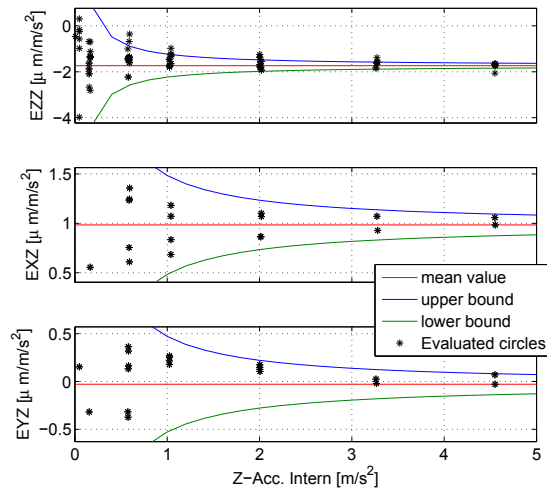


FIGURE 6. Evaluation of Z-Axis with different dynamic settings, direction and planes. Lower dispersion with higher acceleration. Upper/lower bound indicate the dispersion for $\pm 0.5 \mu\text{m}/\text{m}/\text{s}^2$ when using the mean value as reference.

sults of the linear axis of a machine tool at two positions of the workspace. The position of the TCP changes the offset to the guide way and hence the stiffness, which is noticeable in the resulting cross-talk parameter $EX\ddot{Z}$.

TABLE 1. Resulting in-talk (diagonal) and cross-talk values $[\mu\text{m}/\text{m}/\text{s}^2]$ for a machine tool at top of the workspace.

	$\ddot{X}$	$\ddot{Y}$	$\ddot{Z}$
EX	-1.53	-0.11	0.97
EY	-0.16	-3.20	0.10
EZ	0.18	0.18	-1.60

CONCLUSION

The evaluation of the acceleration correlated in-talk and cross-talk errors and their repeatability with higher acceleration concludes the systematic behaviour due to the acceleration state of a machine tool. The difference between external and internal measurements can be correlated with acceleration. In addition, the evaluation method presented here describes an axis-calibration method on quasi-static acceleration level due to low frequent excitation of the measurement motion. The position-dependency of the evaluation has to be considered as the stiffness of the machine struc-

TABLE 2. Resulting in-talk (diagonal) and cross-talk values $[\mu\text{m}/\text{m}/\text{s}^2]$ for a machine tool in the center of the workspace.

	$\ddot{X}$	$\ddot{Y}$	$\ddot{Z}$
EX	-1.34	0.22	1.42
EY	0.22	-3.94	0.14
EZ	0.43	0.19	-1.70

ture changes due to variations of offsets over longer movements. The obtained values are also proposed to be used as feed forward parameters on acceleration level in suitable servo controls.

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DIMENSIONAL METROLOGY OF PLASMONIC NANOLITHOGRAPHY MACHINE

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ABSTRACT

A plasmonic nano lithography machine is being designed to implement the desired motion of flying plasmonic lens [1]. The precision positioning stage for plasmonic lithography requires dimensional metrology for its single degree of freedom motion. Related five degree of freedom errors from primary machine structure components are measured and integrated based on Homogenous Transformation Matrix. A thermal modeling plan is designed to characterize the response of machine due to temperature fluctuation in a room without temperature control, and that of the machine components like optical linear scale for future compensation and correction. Drift test and calibration of interferometer, capacitance gages and thermistors will be provided to validate the error correction in future.

MACHINE STRUCTURE OVERVIEW

The machine under development consists of a single degree of freedom motion with a spinning spindle-disc structure as a whole lithography system in Figure 1.

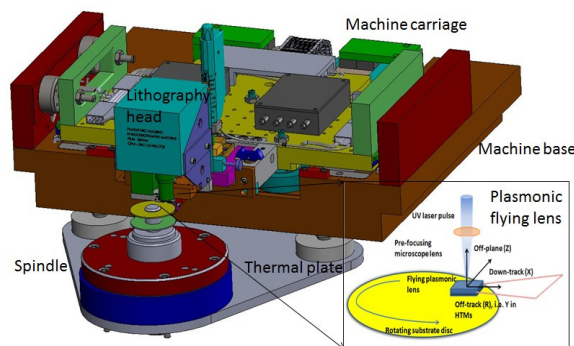


Figure 1 Machine structure and flying plasmonic lens

Driven by two linear motors underneath the carriage, the stage will be moving a flying plasmonic lens 15 nm above a photoresist disc spinning at 2000 RPM in the off-track direction covering 35 mm. Two linear optical scales on left and right sides of the carriage provide the basic

positioning information; one vertical wedge provides fine adjustment motion; and X-Y piezo actuators (3 um) provides final correction and location of a slider attached with the flying head.

GEOMETRIC ERROR MODELLING

Based on the rigid body assumption of machine structures, a single Degree of Freedom (DOF) motion would include six DOF parasitic errors: one linear positioning error, two perpendicular out of straightness errors and three angular errors (pitch, yaw and roll). The metrology loop illustrated in Figure 2 is consisted of base, carriage, wedge and shear PZTs which is closed at the center of spindle. Using HTM representation, a complex model could be simplified to obtain low order error terms and extract those necessary ones to be measured [2-5].

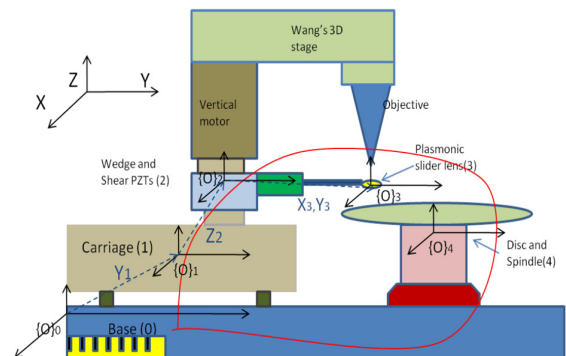


Figure 2 Metrology loop and coordinate system

The error can be written as the difference between commanded machine position and the practically measured machine location in a same coordinate system (i.e. the base system) as below.

$$E_{(0)} = P_{(0)e} - Q_{(0)e} = \begin{bmatrix} \delta_x(Y) + \varepsilon_x(Y) \cdot Z_2 - thermal_d \\ \delta_y(X) + \delta_y(Z_2) - \varepsilon_x(Y) \cdot Z_2 + \alpha_{Z_2} \cdot Z_2 + thermal_{Y_1} - thermal_r \\ \delta_z(Z_2) + \delta_z(Y) - thermal_h \\ 0 \end{bmatrix}$$

In fact, the Z direction deviation perpendicular to the primary Y motion can be neglected because the flying head is designed to follow the profile of spinning disc with a gap of about 10 to 20 nm.

THERMAL ERROR MODELING

Compared with deterministic modeling process, thermal error modeling is based on experimental information [6] collecting from four surface temperature thermistors and one air temperature thermistor, which are located at spindle mount, thermal plate, laser beam and the two linear scales in Figure 3.

All the temperature measurement sensors will be calibrated before the thermal tests. In The expansion of linear scale due to temperature change will be taken into account when software correction on positioning is needed without the measuring laser during the lithography procedure.

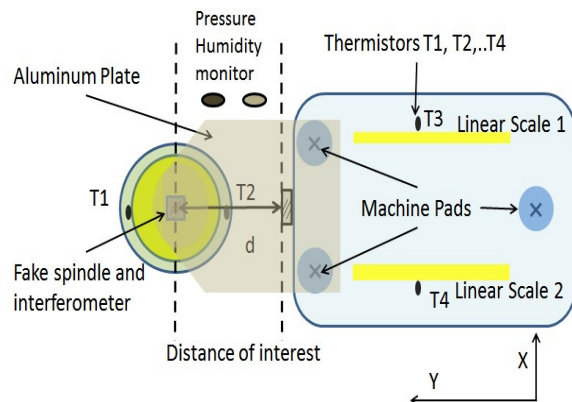


Figure 3 Thermal modeling and sensor location

GEOMETRIC ERROR MEASUREMENT PLAN

To measure all the first order geometric errors in the final HTM representation, a series of measurements will be conducted. In Table 1, different error terms are measured by different instruments. The long range (i.e. larger than 10 mm) measurement task requires the use of laser interferometer which the short range (i.e. less than 1 mm dimensional error and differential angular motion error) ones are measured by using capacitance gage or combination of capacitance gages.

In Figure 4, a plane mirror displacement measuring interferometer is mounted on an adjustment "ring" by using 5 minute epoxy.

Three set screws are used to provide location and alignment without manual touching the optics. A plane mirror is hold by a home designed bracket which also has the capability to align the mirror.

Table 1 Error measurement plan

Type	Symbol	Measurement plan
Linear positioning accuracy	$\delta_Y(Y_1)$	Plane mirror Interferometer (direct)
X straightness of Y of carriage	$\delta_X(Y_1)$	Single cap gage (direct)
Z straightness of Y of carriage	$\delta_Z(Y_1)$	Single cap gage (direct)
Roll angle of Y axis of carriage	$\varepsilon_Y(Y_1)$	Double cap gages (differential manner)
Pitch angle of Y axis of carriage	$\varepsilon_X(Y_1)$	Triple cap gages (combined differential manner)
Squareness of Z and Y of wedge	$\alpha_{Y_2Z_2}$	Observed and estimated by autocollimator (indirect)
Straightness of Z axis of wedge	$\delta_Z(Z_2)$	Integrated with the Z straightness of Y in carriage
Straightness of Y axis of wedge	$\delta_Y(Z_2)$	Integrated with linear positioning accuracy

MECHANICAL PARTS FOR METROLOGY

As an independent metrology system for the lithography machine, a variety of mechanical parts have been designed and machined to provide allocation, alignment and adjustment needs for related sensors, optics and reference artifacts from Figure 4 to Figure 6.

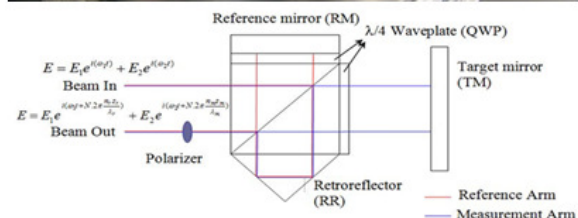
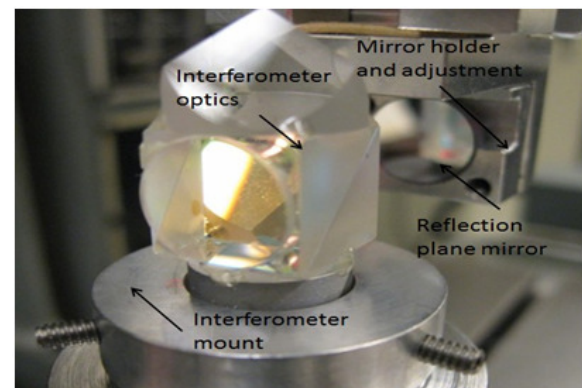


Figure 4 Plane mirror displacement measurement interferometer

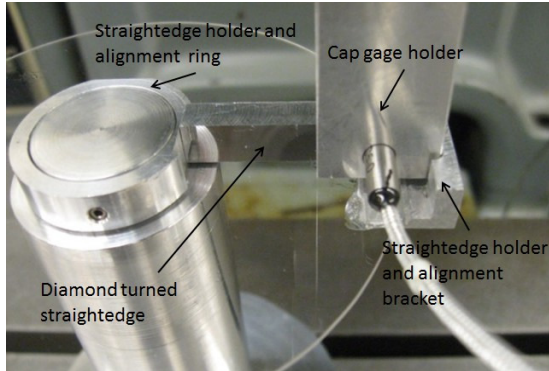


Figure 5 Straightedge reversal measurement setup

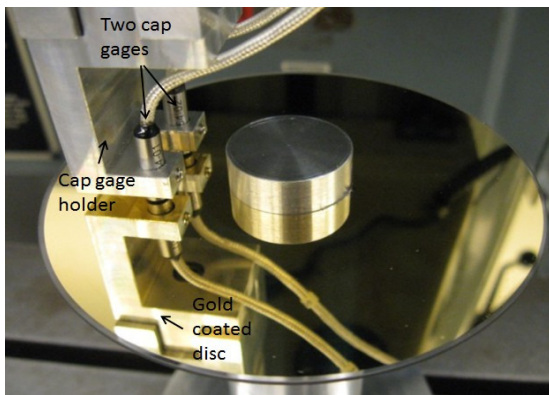


Figure 6 Roll angle measurement setup

CALIBRATION AND COMPENSATION OF METROLOGY UNITS

The capacitance gages were calibrated two year ago at NIST. It's not necessary to redo the calibration again due to our application requirement. Lion Precision C5-D capacitance gage probe is used as proximity sensor. It has a resolution of 6 nm which is a quarter of the 22 nm half pitch lithography resolution.

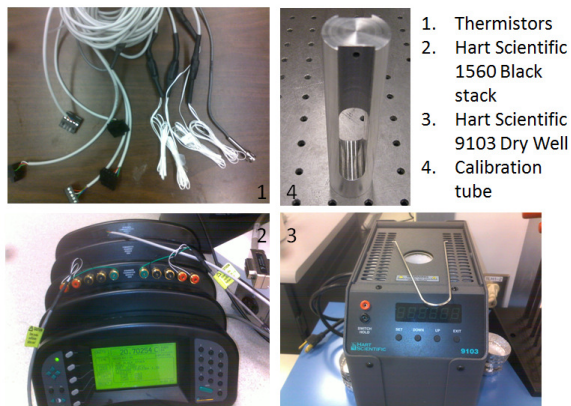


Figure 7 Thermistor calibration system

The temperature sensors have not been calibrated yet. Since they are NTC type semiconductor device i.e. the temperature-resistance coefficients are crucial to certain of temperature measurement range (around 20 degree based on lab condition), we need to build a system to calibrate and obtain those coefficients for extrapolation over a valid period of time.

An independent thermistor calibration system has been designed as illustrated in Figure 7. An Aluminum tube is designed to fit in the constant temperature cavity inside of the dry well with the surface thermistors attached interiorly and isolated from the outside environment. The preset temperature values of dry well will be feed into the Black Stack 1560 system as well as the resistance readings of thermistors through GPIB port.

Although laser used in this study is not need to be frequency stabilized, the machine positioning resolution is 5 nm in without absolute length requirement which requires the correction of measurement due to the environmental factors like temperature, pressure and humidity [7].

SYSTEM LEVEL PERFORMANCE TESTS PROCEDURE AND GOAL

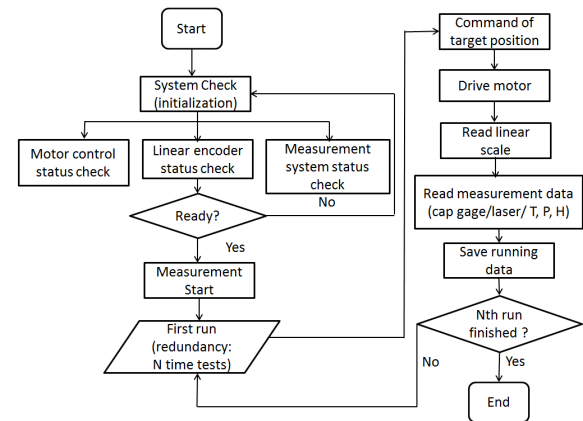


Figure 8 Flow chart of machine calibration procedure

The procedure of machine calibration can be described by Figure 8, where a flow chart illustrates the whole process from initialization of driving mechanism, machine encoder and measurement system to the process of stage motion and parallel data acquisition including environment and laser interferometry then to the end of a whole running.

Redundant tests are needed to check the repeatability of the systematic error and thermal performance. The thermal test is an

independent process in which the whole machine will be located in a lab room whose temperature can be set up covering up to ten degree Celsius. Then all the tests in the flow chart will be repeated to check the response of laser, machine error in terms of machine thermal behavior.

In Figure 9, a block diagram illustrates the communication layout of the calibration system. The signal from Renishaw linear optical scales will be used as measurement triggers. Both pressure and temperature measurement information will be feed to computer through GPIB cable in the LabVIEW platform. The interferometer system requires an USB expansion module to achieve laser readings. And the cap gage signal will be directly feed in the computer by NI PCI card.

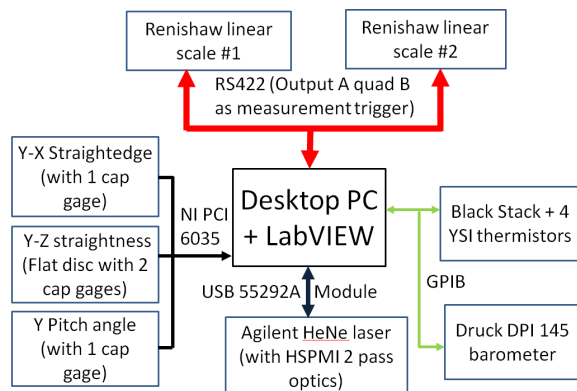


Figure 9 Block diagram of measurement system

FUTURE WORK

The developed setups will be tested under LabVIEW environment. An error table containing correction values will be generated and integrated with control algorithm. The repeatability of both geometric and thermal errors will be studied to validate the effectiveness of software correction.

ACKNOWLEDGEMENT

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Browy contributed to the diamond turning of the straightedge.

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NON-CONTACT PROBE FOR ON-MACHINE METROLOGY

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INTRODUCTION

On-machine metrology is particularly important for diamond turning and grinding as it is difficult to remount and align a part if it does not meet off-line inspection criteria. There is also the issue of tool wear; a process that started well may fail part way through the cut, and if tool replacement is needed, it is vital to know that before removing the part. A means of rapid, non-contact, *in situ* profiling and roughness measurement could improve the productivity of diamond tool machining.

Recently we first showed that diamond turning machines are sufficiently isolated that steady fringes can be obtained by simply setting a Point Source Microscope [1,2] equipped with an interferometric Mirau objective on the cross slide of a machine. Further, we demonstrated that the machine can be precisely driven to get temporally shifted fringes so that common algorithms can be used to obtain area based surface roughness measurements. This led to the question of whether essentially the same hardware could be used to rapidly profile diamond turned parts. We show via simulation that the answer is yes and that the approach can be implemented rather simply.

We first describe the PSM and its configuration as a Microfinish Topographer (MFT) by using interferometric data reduction software. Then we describe how this hardware is changed into a profiler by changing the light source and camera. Finally, we show how this hardware that we call a Non-Contact Profiler [3] (NCP) is used on a diamond turning machine to profile turned or ground parts *in situ*.

MICROROUGHNESS TOPOGRAPHER (MFT)

The Point Source Microscope [1] has been described previously while the MicroFinish Topographer [2] (MFT) is the same hardware where the LED illumination channel is used as the light source and a Mirau interference

objective replaces the standard infinite conjugate objective as shown in Fig. 1.

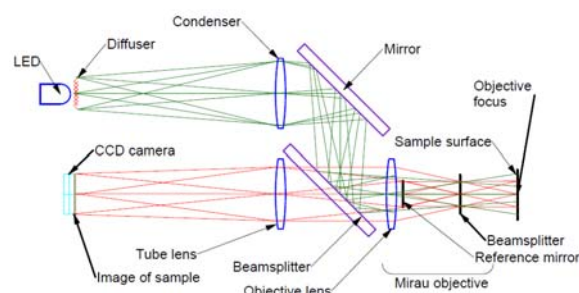


FIGURE 1. Ray paths in the MFT for illuminating and imaging the sample surface.

The red LED source has sufficient coherence to obtain interference fringes within about $\pm 8 \mu\text{m}$ of the reflection from the reference mirror in a 10x Mirau objective. This is more than enough coherence to get surface roughness data over a $1 \times 0.75 \text{ mm}$ area with a 10x objective as long as the objective reference mirror is within about 1° of the normal to the surface. Higher magnification objectives have proportionally smaller fields of view but can accommodate greater tilt between surface and reference.

Commercially available software drives a PZT behind the objective to shift the fringes to produce roughness data. Alternatively, machine control based software can step the MFT in $1/4$ fringe intervals to capture 4 or more interferograms, and reduce the data to provide surface roughness information.

WHITE FRINGE LOCATION

Quite separately from our work on roughness metrology we found that by using a white light LED and a color camera, it is easy to obtain three fringe intensity patterns, one for each color as shown in Fig. 2. When the phases of all three fringe intensity patterns are aligned we get a white or black fringe depending on whether the

surface is a dielectric or metal, respectively, indicating that the surface and reference mirror in the objective are precisely conjugate, or the same distance from the beamsplitter. In white light scanning interferometry this is done by looking for a peak in fringe contrast. With the 3 distinct wavelengths in a white LED there are 3 distinct fringe intensity patterns captured by the color camera. On the left of Fig. 2 is the white LED fringe pattern as seen using a 10x Mirau objective and a color camera, while on the right is a plot of the three intensities from one row of the data. The phases of all three sinusoidal intensity patterns from a row of data align at the location of the white fringe as indicated by the black line tying the figures together.

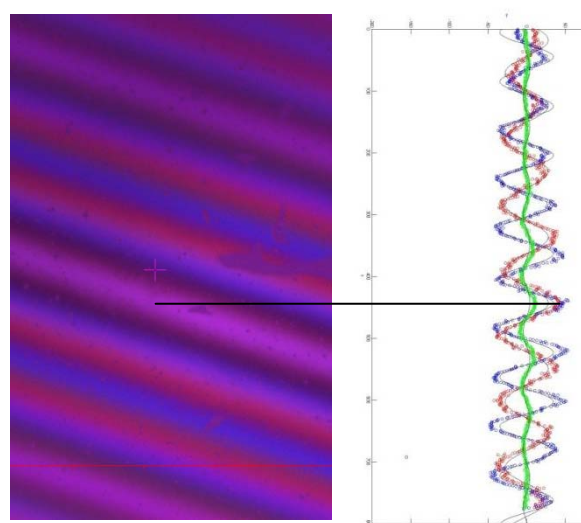


FIGURE 2. Fringe pattern obtained with a white LED and a color camera (left), intensities of the three colors where the phases align in a column of pixels at the end on the black line (right)

The realization that we could easily recognize the white fringe using the same hardware as the MFT led directly to the idea that the same device used for roughness measurement could be used as an indicator, or sensor, to profile based on the location of the white fringe in the frame of data. We call the MFT used in this way to profile the Non-Contact Profiler (NCP).

INFORMATION IN THE WHITE FRINGE

The information for contouring comes out of the algorithm used to find where the phases align. A sine wave is fit to each of the three intensities in the form $I = A \sin(B \cdot x + C)$. The amplitude A is of no interest while B is proportional to the number of fringes in a frame of data, or the tilt

between the reference mirror and the patch of surface observed. For a given frame of fringes, B is also inversely proportional to the wavelength being fit because there will be more fringes for a shorter wavelength. The white fringe is at the value of x (in pixels) where the argument $(B \cdot x + C)$ is equal $N\pi/2$ for odd integral N . Once B and C are found for each row (or column) of intensity data, the y value (or x value), in pixels, is known for the position where the reference mirror and surface are conjugate, or the same distance from the beamsplitter in absolute terms.

Fig. 3 shows an example of a 2 μ inch (50 nm) rms surface from a GAR surface roughness standard artifact [4] with the x,y locations of the center of the white fringe indicated with a row of dots. The scale in the Figure was set by the 10x objective; the optical path difference between fringes for the blue wavelength is about 240 nm and in this example there are about 100 pixels between fringes. Once the fitting is done to find the white fringe, and assuming there are no big jumps from row to row of data, the center of the fringe can be found with a simple algorithm that takes less than 1 ms per row. This means surfaces can be profiled with 1 μ m spatial resolution at a rate of 1 mm/sec and proportionally faster at lower spatial resolutions and with height resolutions of a few nm rms.

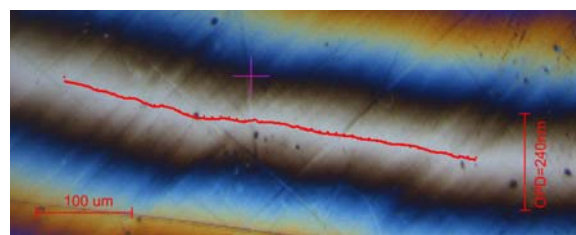


FIGURE 3. White fringe showing profile of a 50 nm rms surface using a 10x Mirau objective

One mode of using the NCP is as an optical micrometer. For example, in Fig. 3 if one wanted to know how far to move the NCP in z to bring the center of the white fringe to the cross, the column of data containing the cross would be fit to find B and C . The C coefficient tells how far to move in nm scaled by the fringe spacing B in pixels per nm.

This mode could be used to find spindle growth by laterally positioning the cross at the vertex of

a part and then moving in z to set the white fringe on the vertex. After running the spindle for a time, the movement of the white fringe from the cross gives the spindle growth. Similarly, in the indicator mode, parts could be squared with the face plate.

The usefulness of this mode is that the height or angle of a particular feature in the field of view can be compared to another to a few nm rms or a few seconds of arc for nm or second differences. The sensitivity is proportionally less the greater the difference in feature height or angle unless the diamond turning machine is used to re-position the NCP. If the NCP is repositioned to have just a few fringes in the field of view, the sensitivity is again in the few nm or second regime. Said in another way, the NCP has high sensitivity when used in a null detector mode but features of interest can be selected on a μm scale spatially.

USING THE NCP AS A PROFILER

Using the NCP as a profiler on a diamond turning machine for near flat surfaces means mounting the NCP on the x slide with set and forget adjustments to center the field of view on the spindle axis and a few minutes of arc off normal so there are a few near vertical fringes across the detector. As long as the angle between reference mirror and surface is no more than about 1° for a 10x objective, the white fringe can be kept in its initial position as the x axis is scanned by using the value of C as a feedback signal to the z axis. A safer and more practical mode would be to run the theoretical part curve and record C as a function of the x axis position to create a contour error map.

For parts with a steep enough radius to have normals greater than 1° when using a 10x objective, the NCP must be mounted on a B axis rotary table and the B axis used to keep the NCP near normal to the surface as it is scanned in x and z as illustrated in Fig. 4. Again, the most practical method of contouring is to run the part contour path and watch for changes in the white fringe location as the indication of errors in the part.

The distance between the objective focus with the white fringe in the middle of the display and the B axis is the "tool radius" used in calculating the measurement profile. To go from cutting the part to measuring is a matter of changing from the "tool radius" to the "measurement radius"

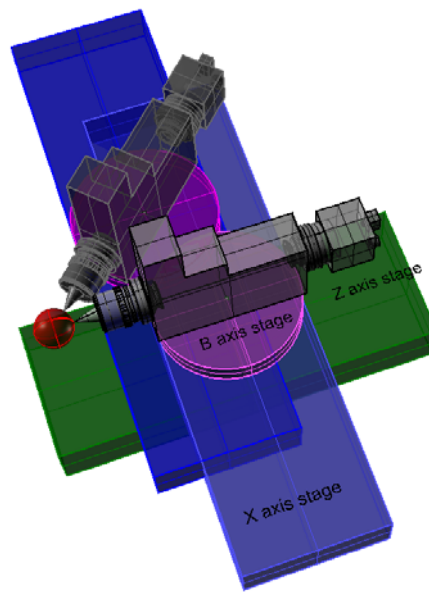


FIGURE 4. NCP mounted on a B axis profiling a ball as the x axis moves from the ball vertex

where the measurement radius is pre-calibrated and stored in the machine control software as a new "tool radius". Clearly, errors in the measurement of the tool radius will lead to errors in the measured contour. However, calibrating the tool radius for measurement purposes is easily done by contouring a good grade steel ball.

Radial errors in the B axis will contribute directly to errors in the measurement of the part contour. However, the oil hydrostatic B axis bearing can be compensated to under 20 nm. Errors in all three angular degrees of freedom in the B axis rotation and axial translation of the B axis make only second order contributions to measurement errors.

Although we have not yet profiled a part on a diamond turning machine, we have simulated real fringe data by thermally changing the optical path between a diamond turned part and the objective reference mirror and shown that the white fringe center can be tracked at 30 frames per second. We have also shown that fringe centers within each frame of data can be found at least as fast as 1 msecond per pixel row or column of data in the frame. This translates to a spatial resolution of $1 \mu\text{m}$ on the surface using a 10x Mirau objective.

FUTURE WORK

Clearly, our next move is to mount the NCP on a machine and profile a known part. We hope to do this in the next couple of months to obtain data showing *in situ* profiling performance. In addition, we want to profile a non-rotationally symmetric part. If, using a 10x objective, the normal in the y direction is less than 1°, we can do this profiling with the setup just as described including profiling while the part is rotating on the spindle at low rpm.

Further, surface roughness measurements can be made at any point in the process of profiling by moving in steps normal to the surface. Roughness data can be obtained at angles as great as 1° for a 10x objective as well since the diamond turning machine is well isolated and there is little danger of fringe print through. If the normals are greater than 1° we can go to higher power objectives to get up to 5° off normal with a 50x objective and smaller field of view.

For still greater angles we will have to add an A axis of rotation and use the white fringe position from both row and column data to produce profiles in both the radial and tangential directions. In this case the initial tilt of the NCP to the surface under test would be set to produce fringes at roughly 45° to the machine coordinate system. Further, the profiling could be done on a spiral basis to give areal coverage of non-symmetric and free-form parts.

CONCLUSIONS

We have described a simple method of performing *in situ*, non-contact profiling of diamond turned parts using a well proven sensor head with a pseudo-white light source and a color camera instead of the normally used red LED and monochrome camera. The color source and camera provide a means of unambiguously isolating the white, or black, fringe that shows precise conjugation of the reference with the surface under test. This permits making profile measurements with a precision of a few nm rms and a spatial resolution of a few μm while taking data at a contouring speed of about 10 mm/second.

For surfaces that depart from plane by 1° or less, all that is required for profiling is a PSM with a Mirau objective, a white light LED source and fairly simple algorithms. This same mode of operation works for angles up to 5° by using a

50x objective although proportionally smaller fields of view are seen.

For symmetric surfaces with larger slopes, the sensor head must be mounted on a B axis to keep the number of fringes from becoming too great to resolve. For steep non-symmetric surfaces, an A axis must also be added. In addition to this profiling capability, there is the simultaneous ability to measure surface roughness by stopping the scan and obtaining 4 or more interferograms at the desired location. No additional alignment is necessary because the NCP is already near normal to the surface, and the machine motion control can do the phase stepping.

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EXPERIMENTAL CHARACTERIZATION OF A FLEXIBLE CAPACITIVE PRESSURE TRANSDUCER

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INTRODUCTION

This paper presents an experimental characterization of a flexible pressure transducer towards understanding its suitability to a manufacturing process control application. Flexible pressure transducers made from soft materials (e.g. low modulus elastomers) have been under development for applications in soft robotics and wearable electronics. The approaches reported in literature have incorporated conductive elements into soft elastomers (e.g. polydimethylsiloxane / PDMS) to form either resistive [1, 2, 3] or capacitive [4, 5] transducers that convert an applied force to an electrical signal using the mechanistic behavior of the elastomeric body.

A possible industrial application of flexible pressure transducers is in roll based lithography. Studies have shown that processes like nanoimprint lithography [6] and microcontact printing [7] are sensitive to the contact pressure distribution between the substrate and the patterned printing roll. A flexible pressure transducer can be incorporated into either the tool (in the case of microcontact printing) or a backup roll (in the case of nanoimprint lithography) to provide an *in situ* measure of contact pressure. This approach would enable both process monitoring and process feedback control.

This study examines a new type of flexible pressure transducer inspired by elastomeric stamps used in soft lithography, similar in operation to that reported by Mannsfeld et al [5]. In our device, a conductive layer is encapsulated within an elastomeric stamp that contains a series of patterned microfeatures (Figure 1a). As pressure is applied between the flexible transducer and a rigid ground plane, the patterned features elastically deform to reduce the height of the stamp conductor (Figure 1b). The capacitance between

the stamp conductive plane and the rigid ground plane can be measured to determine the pressure and displacement imposed on the features. This device has the potential for high resolution measurement of local contact pressures.

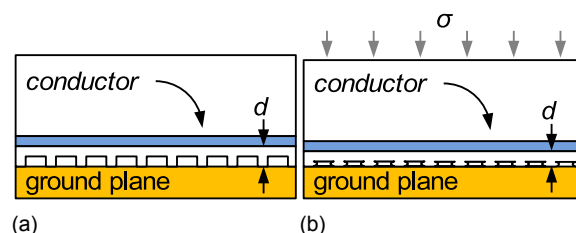


FIGURE 1. A pressure transducer is formed by encapsulating a conductive layer in an elastomeric stamp with microfeatures that are placed on a ground plane (a). When an external pressure is applied (b), the features elastically deform to alter the capacitive gap d between the stamp conductor and ground plane.

THEORY OF OPERATION

Capacitance is inversely proportional to the displacement between parallel plates. Stress transduction in this sensor is thus dependent on the mechanical load-displacement behavior of the deformable microfeatures.

Small static deformation of PDMS microfeatures is well understood from applications like soft lithography [8]. In a precision sensor application, two additional phenomena are important: nonlinear large deformation behavior and time dependent viscoelastic behavior.

While elastomeric microfeatures deform linearly at small displacements, large displacements lead to nonlinear behavior from both large deformation kinematics and hyperelastic stiffening. These effects combine to produce an inherently nonlinear

constitutive relationship between stress and deformation (and hence capacitance).

Elastomers are viscoelastic materials, where both creep and stress relaxation must be considered. At short timescales, creep acts to dampen material deformation, effectively a low pass filter that limits ultimate sensor bandwidth. At longer timescales, stress relaxation behaves as a high pass filter to limit the achievable sensor accuracy.

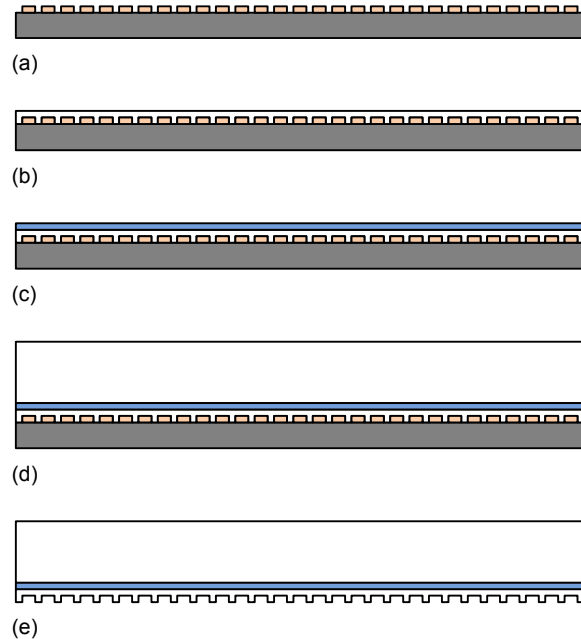


FIGURE 2. Cross sectional schematic of sensor fabrication. (a) PDMS is deposited against a patterned wafer using spin coating. (b) A conductive polymer layer is deposited on the PDMS using spin coating. (c) A thick layer of PDMS is cast using injection molding. (d) The entire assembly is parted from the wafer, resulting in an elastomeric sensor.

DEVICE FABRICATION

The flexible pressure transducer was fabricated using microfabrication techniques as illustrated in Figure 2. A 100 mm silicon wafer was patterned with SU8 2005 (Microchem) to produce a hexagonal pattern of 5 μm features. The patterned wafer was treated with hexamethyldisilazane to prevent adhesion of the subsequent polymer layers. PDMS (Dow Corning Sylgard 184) was applied to the wafer, degassed in a vacuum desiccator, and thinned by spin coating. After curing the PDMS, a layer of conductive polymer (PEDOT:PSS, Heraeus Clevios S V3 HV) was applied by spin coating and annealed. Finally, a

thick layer of PDMS was cast against the wafer by injection molding.

Removing the polymer layers from the patterned wafer produced a microfeatured PDMS slab with an encapsulated layer of conductive polymer.

EXPERIMENT

Experiments were conducted by mounting a 20 mm square sample (400 mm^2 area) of the flexible sensor against a printed circuit board (PCB) and cycling the construct in a load frame. Electrical connection between the PCB and stamp conductor was made using a droplet of liquid PEDOT:PSS solution (Figure 3).

The sensor capacitance ($C_0 = 2 \text{ nF}$) was measured using an resistor-capacitor (RC) low pass circuit. The circuit output was measured with a RMS to DC converter and recorded with a National Instruments PCI-6220 data acquisition card with a 1 kHz hardware antialiasing filter.

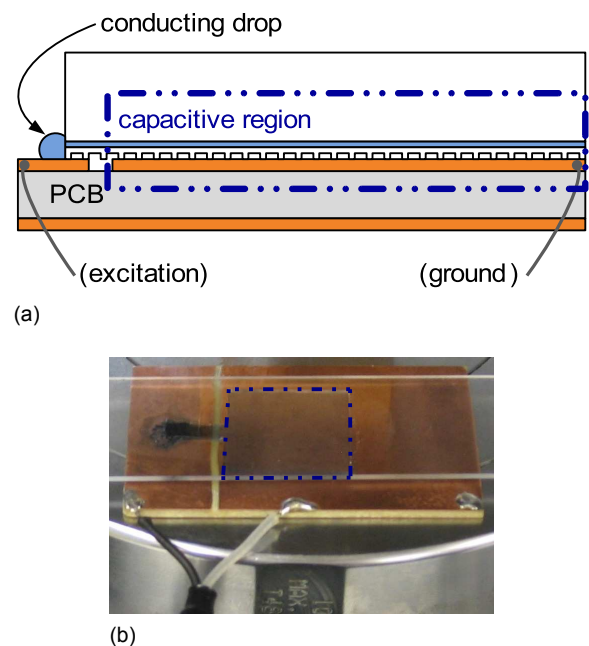


FIGURE 3. (a) Schematic of capacitor construction and (b) photograph of experimental implementation. A copper PCB is divided into two separate regions by etching a single trench. The elastomeric sensor is laminated against the PCB and electrically connected to one half of the PCB using a droplet of conductive polymer. The resulting parallel plate capacitor area is outlined in each figure.

RESULTS

The sensor noise was quantified using 1000 s of RC circuit output data recorded at a 40 kHz sampling rate. The resulting power spectral density is shown in Figure 4. These data show that the sensor has a resolution (above 1 Hz) of $400 \mu\text{V}$ (25 Pa) and an accuracy (below 1 Hz) of 8.6 mV (500 Pa). The sensor tested has a useful output range of about 0.5 V / 10 V, giving a sensor dynamic range of 62 dB.

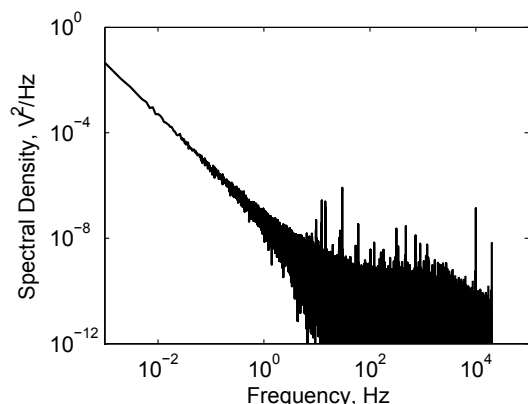


FIGURE 4. Power spectral density of sensor noise. The flicker / Johnson noise cutoff is at 10 Hz and $10^{-9} \text{ V}^2/\text{Hz}$, with high frequency rolloff from the antialiasing filter at 1 kHz.

A typical experimental impulse response is shown in Figure 5, generated with small strikes to the sensor. This impulse response shows about a 16 ms time constant during the decay, corresponding to a 10 Hz sensor bandwidth.

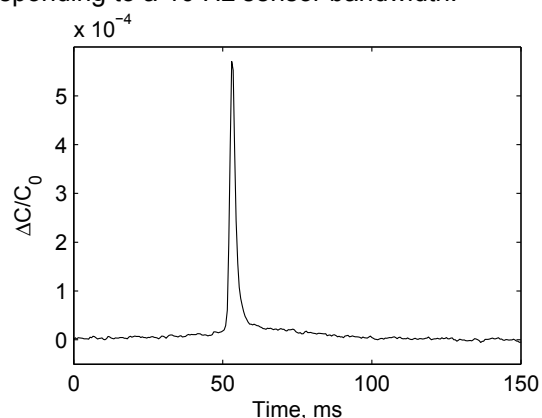
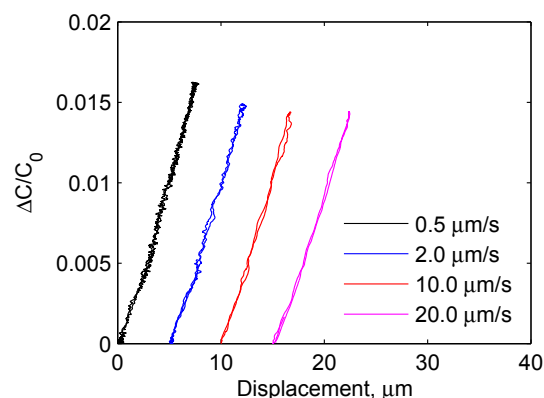


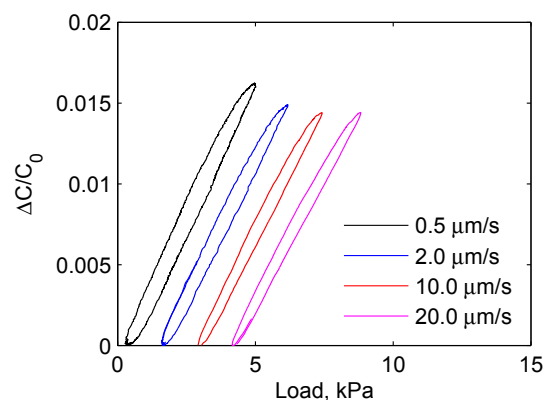
FIGURE 5. Sensor response to a pressure impulse, showing a noticeable decay period of about 50 ms.

The linearity and hysteresis of the sensor were characterized through cyclic loading at different strain rates and load maxima.

Figure 6 shows the behavior of the sensor loaded at different strain rates between 0 and 5 kPa. At these small displacements, the displacement-capacitance relationship is linear within 5 % and exhibits no detectable hysteresis. The load-capacitance data indicates linearity but exhibits 10% hysteresis that does not have a strong correlation to strain rate.



(a)



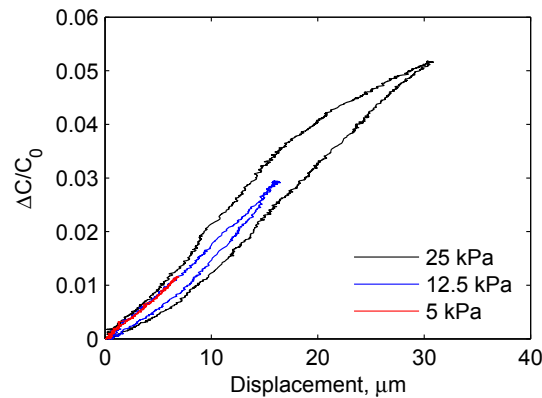
(b)

FIGURE 6. Small signal sensor output during cyclical loading as a function of strain rate. Performance appears linear but the load-capacitance relationship exhibits rate independent hysteresis.

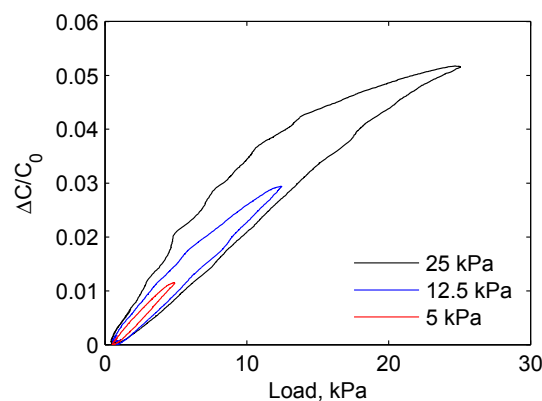
Figure 7 shows the sensor behavior at different cycling limits. While the sensor behavior appeared linear over small displacements, larger displacements result in both (i) nonlinear behavior and (ii) increased hysteresis.

DISCUSSION AND CONCLUSION

These experimental results highlight characteristics of elastomeric pressure transducers that must be considered during system design. While the sensors can achieve adequate resolution and dy-



(a)



(b)

FIGURE 7. Sensor output over different ranges of excitation. Over large ranges, the sensor behaves in a nonlinear fashion with increasing hysteresis. The load-capacitance data has hysteresis of 20%, 12%, and 9% at each respective excitation range (25 kN, 12.5 kN, and 5 kN).

namic range, their accuracy is limited by effects of stress relaxation and hysteresis. Material damping may limit accuracy at bandwidths above about 10 Hz.

Despite these limitations, it is believed that these sensors can be deployed in an industrial setting like *in situ* pressure sensing in roll to roll printing. A complete dynamic model of sensor performance, coupled with the inherently cyclic nature of roll based processing, should provide sufficiently accurate sensor performance.

ACKNOWLEDGMENTS

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PERFORMANCE EVALUATION OF DATA FUSION METRICS

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ABSTRACT

Typically, well-recognized images like 'Lena' are induced with single-scale noise to compare the performance of fusion metrics. But surface metrology datasets are distinct as all the data points are in focus and also some engineered surfaces exhibit fractal or multi-scale nature. So, a performance study was conducted on datasets obtained from a wide range of surface samples. A new methodology was used for inducing multi-scale noise. Based on the study, it was shown that Mutual Information (MI), Universal Quality Index (UQI) and Structural Similarity Index (SSIM) are the preferred fusion metrics for surface metrology datasets.

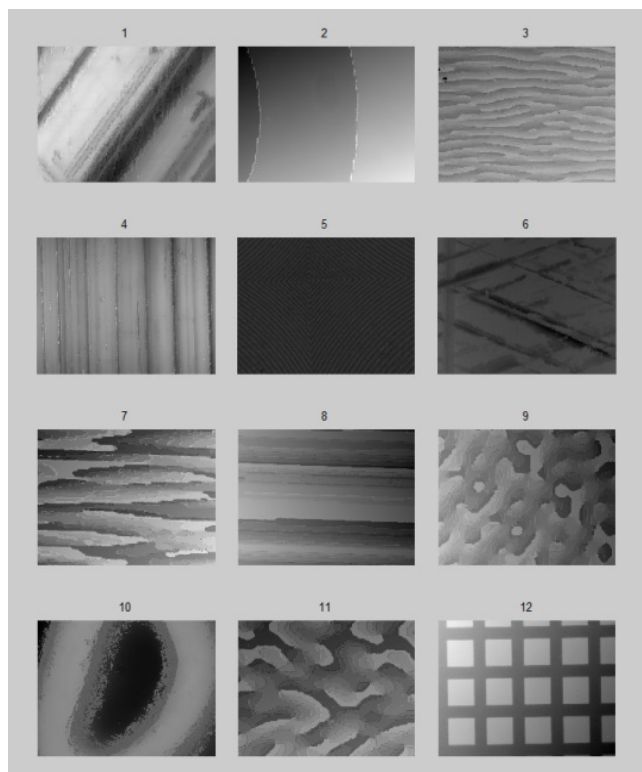


FIGURE 1: Sample datasets chosen for the comparison study (1) Lapped surface (2) beam shaper optical surface (3) Dipole diffuser optical surface (4) Turned surface (5) Fresnel lens surface (6) Honed surface (7),(8) and (9) Pattern generator surfaces (10) Formed surface (11) Spot array generator (12) Square grid array

BACKGROUND

Standard images like 'Lena or Lenna' are used to conduct the performance study. But typical engineered surface datasets are obtained with 'infinite focus'. Hence there was a need to evaluate the performance of the fusion metrics for the surface metrology domain. The study was performed on twelve datasets shown in figure 1 covering a wide range of structured and unstructured engineered surfaces. Each dataset was treated with ten noise conditions - Gaussian noise with three variance levels (0.005, 0.01 and 0.015), and Speckle noise with three variance levels (0.02, 0.04 and 0.06), Salt & Pepper noise with three noise density levels (0.025, 0.05, and 0.075), and Poisson noise.

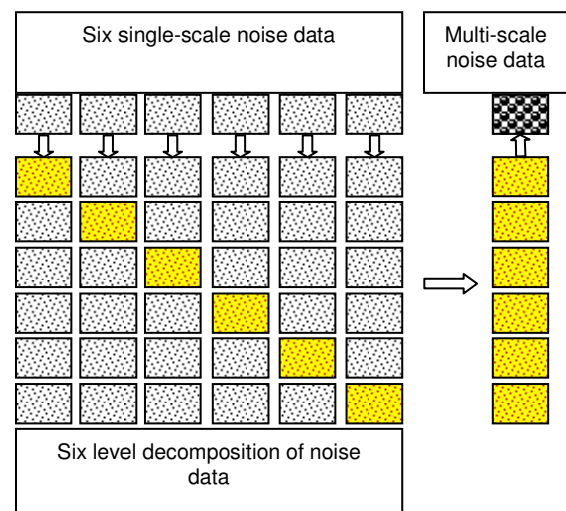


FIGURE 2: Schematic showing the method of generation of multi scale noise.

Traditionally, noise is induced at single-scale. In this paper, a new methodology (shown in figure 2) for generating multi-scale noise is proposed and used to compare the fusion metrics performance. A set of six noise maps are generated for the individual noise type and a six-level Discrete Wavelet Frame (DWF) is performed with B3 spline using 3D Hybrid method [1]. A sub image from each noise map is taken to generate a new multi-scale noise map. Figure 3 shows the original data obtained

from a lapped surface and data sets after adding different types of multi-scale noise.

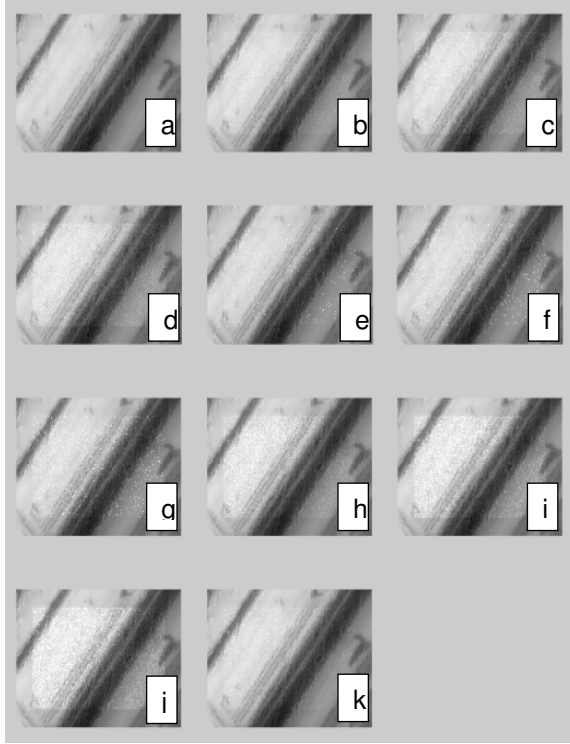


FIGURE 3: (a) Original Image (b)-(d) Image obtained after addition of multi-scale Gaussian noise (e)-(g) Images obtained after addition of multi-scale salt and pepper noise (h)-(j) Images after addition of multi-scale speckle noise (k) Image after addition of Poisson noise.

EVALUATION OF FUSION METRICS

In order to characterize the performance of individual metrics, set of twelve datasets and eleven noisy versions of those datasets were evaluated using fusion metrics Root Mean Square Error (RMSE), Peak Signal to Noise Ratio (PSNR), Cross Entropy (CE), Mutual Information (MI) [2], Universal Quality Index (UQI) and Structural Similarity Index (SSIM) and Multi-Scale Structural Similarity Index (MS-SSIM).

- **Root Mean Square Error (RMSE):**

$$RMSE = \sqrt{\frac{1}{NM} \sum_{i=1}^N \sum_{j=1}^M |R(i, j) - F(i, j)|^2}, \text{ where } R$$

denotes reference data, F denotes fused data, (i, j) a given data point and $N \times M$ is the size of the data. $RMSE$ increases with the increase in deviation between R and F . The method which results in least $RMSE$ value is preferable.

- **Peak Signal to Noise Ratio (PSNR):**

$$PSNR = 10 \log_{10} \left(\frac{L^2}{\frac{1}{NM} \sum_{i=1}^N \sum_{j=1}^M |R(i, j) - F(i, j)|^2} \right), \text{ where}$$

L is the number of gray levels in the image, after the data is converted to a normalized gray scale image d based on the heights in the data. $PSNR$ will be infinity when R and F are exactly same. The data fusion process that yields the highest $PSNR$ value is desirable.

- **Mutual Information (MI):**

$$MI = \sum_{i_1=1}^L \sum_{i_2=1}^L h_{R,F}(i_1, i_2) \log_2 \frac{h_{R,F}(i_1, i_2)}{h_R(i_1)h_F(i_2)}, \text{ where } h_{R,F}$$

denotes the normalized joint gray level histogram images R and F , obtained by converting the datasets into gray scale images and h_R , h_F are the normalized marginal histograms of the two images.

- **Universal Quality Index (UQI)[3]:**

$$UQI = \frac{4\sigma_{xy}\bar{x}\bar{y}}{(\sigma_x^2 + \sigma_y^2)[(\bar{x})^2 + (\bar{y})^2]}, \text{ where } x \text{ and } y$$

denotes the reference and test images obtained by converting the datasets into gray scale images and

$$\bar{x} = \frac{1}{N} \sum_{i=1}^N x_i, \quad \bar{y} = \frac{1}{N} \sum_{i=1}^N y_i, \quad \sigma_x^2 = \frac{1}{N-1} \sum_{i=1}^N (x_i - \bar{x})^2$$

$$\sigma_y^2 = \frac{1}{N-1} \sum_{i=1}^N (y_i - \bar{y})^2, \quad \sigma_{xy} = \frac{1}{N-1} \sum_{i=1}^N (x_i - \bar{x})(y_i - \bar{y})$$

UQI calculates the amount of salient information that has been effectively transferred from image x to image y . The possible values for UQI range from -1 to 1 and the best value 1 would be achieved images x and y are exact.

- **Overall Cross Entropy (CE):**

$$CE(X, Y; F) = \frac{CE(X; F) + CE(Y; F)}{2}, \text{ where}$$

$$CE(X; F) = \sum_{i=0}^L h_X(i) \log_2 \frac{h_X(i)}{h_F(i)} \text{ and } X, Y \text{ are the}$$

source images and F is the fused image, obtained by converting the datasets into gray scale images

- **Multi Scale Structural Similarity Index (MS-SSIM) [4,5]**

$$MS-SSIM(X, Y) = [l_M(X, Y)]^{2\alpha_M} \cdot \prod_{j=1}^M [c_j(X, Y)]^{\beta_j} [s_j(X, Y)]^{\gamma_j}$$

where, one of the original datasets X and the fused dataset Y are taken and both are decomposed to M levels. $l_M(X, Y)$ is the

luminance comparison factor, which is computed only at the largest scale M and $c(X,Y)$ and $s(X,Y)$ are the contrast and structural similarity comparison factors computed at all scales.

$$l(X,Y) = \frac{2\mu_x\mu_y + C_1}{\mu_x^2 + \mu_y^2 + C_1} \quad c(X,Y) = \frac{2\sigma_x\sigma_y + C_2}{\sigma_x^2 + \sigma_y^2 + C_2}$$

$$s(X,Y) = \frac{\sigma_{xy} + C_3}{\sigma_x\sigma_y + C_3}, \text{ where } \mu_x, \sigma_x \text{ and } \sigma_{xy} \text{ are}$$

mean of X , standard deviation of X and covariance of X and Y respectively and C_1 , C_2 and C_3 are small constants given by $C_1 = (K_1L)^2$, $C_2 = (K_2L)^2$ and $C_3 = C_2/2$, where L is the dynamic range of the gray scales (255), $K_1 \ll 1$ and $K_2 \ll 1$. α , β , γ are parameters chosen according to the importance of three factors – luminance, contrast and structural similarity. When $\alpha = \beta = \gamma = 1$, and $C_1 = C_2 = 0$, the structural similarity index results in universal image quality index.

With the known levels of noise that was added to the data, the expected quality level should closely track the noise levels demonstrating a specific trend, as shown in figure 3.

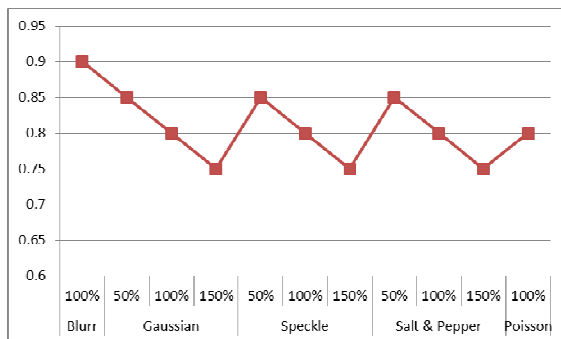


FIGURE 3: Expected quality metric to correspond to the levels of additive noise.

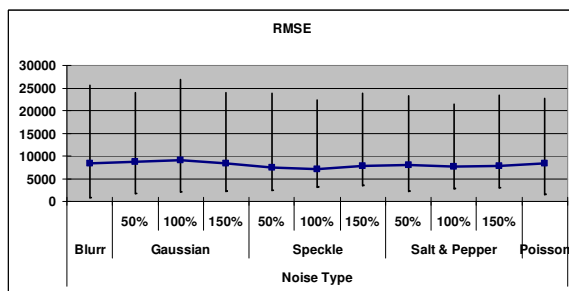


FIGURE 4: Root Mean Square Error (RMSE) values obtained on 12 datasets with 11 noise levels.

RMSE (shown in figure 4) was able to differentiate between noise levels within each

data type, but there was no correlation between data types, as the metric doesn't follow the

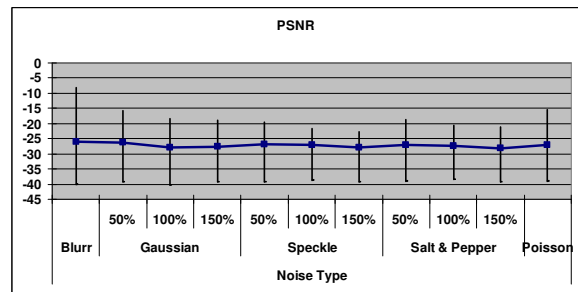


FIGURE 5: Picture Signal to Noise Ratio (PSNR) values obtained on 12 datasets with 11 noise levels.

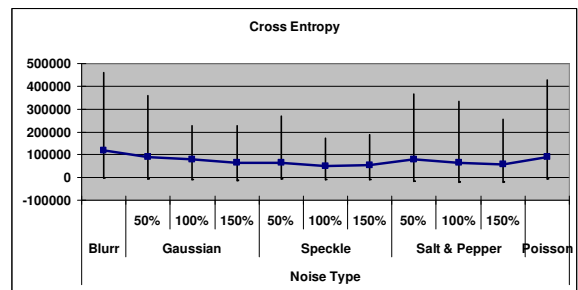


FIGURE 6: Cross Entropy (CE) values obtained on 12 datasets with 11 noise levels.

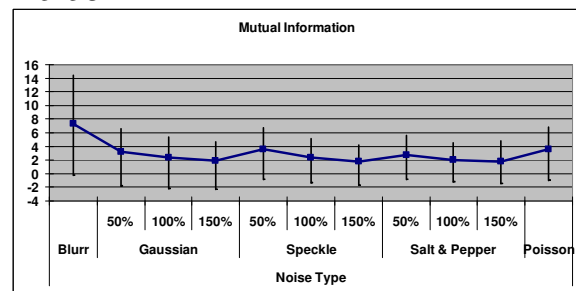


FIGURE 7: Mutual Information (MI) values obtained on 12 datasets with 11 noise levels.

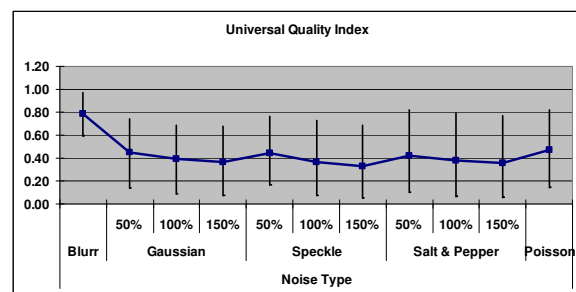


FIGURE 8: Universal Quality Index (UQI) values obtained on 12 datasets with 11 noise levels.

expected trend. PSNR and CE performed similarly, as shown in figure 5 and 6 respectively.

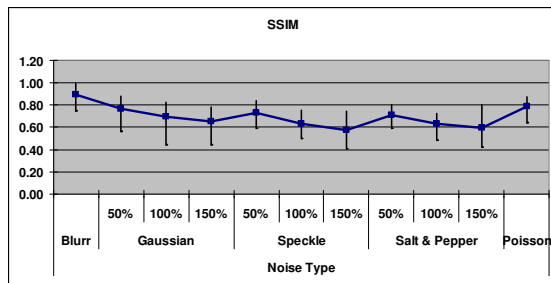


FIGURE 9: *Structural Similarity Index (SSIM) values obtained on 12 datasets with 11 noise levels.*

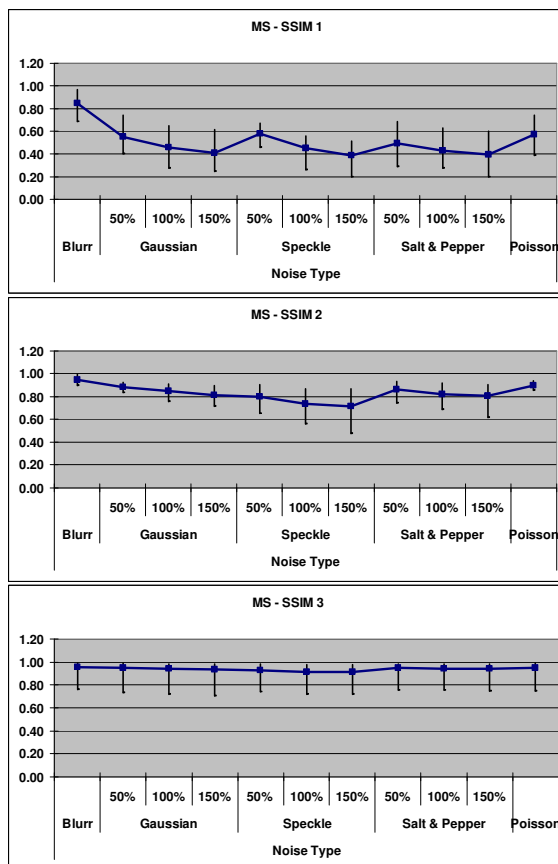


FIGURE 10: *Structural Similarity Index (SSIM) values obtained on 12 datasets with 11 noise levels, for first three levels.*

MI, UQI and SSIM (shown in figure 7, 8 and 9 respectively) were able to differentiate between noise levels as well as show a stable correlation on all data types, similar to the expectation shown in figure 3.

Figure 10 shows the calculated SSIM values for first three sub-images. Even though MI was able to differentiate the impact of noise on 'Data no.5' similar to MS-SSIM, the potential drawback to this metric is that it doesn't provide a baseline to evaluate like UQI, SSIM and MS-SSIM, which are set to have a maximum value of 1.

CONCLUSION

Based on the performance in this study, MI, UQI, and SSIM were recommended as preferred metrics for single-scale based performance metrics. MS-SSIM is recommended as the preferred metric for multi-scale based performance.

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Density Measurement of Tantara (Ta_2O_5) Foams Using X-Ray Radiography

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Abstract

Successful High Energy Density Science experiments at the National Ignition Facility (NIF) require that millimeter-scale targets are fabricated, aligned, and measured with micrometer-scale tolerances. In addition to the stringent dimensional requirements, inherent bulk properties such as density need to be understood to the %1.0 level. LLNL has used the combination of dimensional metrology, chemical analysis and x-ray radiography to meet this specification for Tantara (Ta_2O_5) foams.

In the case of targets containing low density materials, such as 500 mg/cc Tantara (Ta_2O_5) being discussed in this paper, density variation due to the bulk material manufacturing and/or machining processes is a critical element of the experiment. In general, density variations such as voids or inclusions can occur during material synthesis or during machining operations and often have the most direct impact to the experiment. However, contaminants like absorbed and adsorbed moisture from the environments seen by the sample as it goes through a number of process steps can also have a large effect on the experiment depending on the material chemistry.

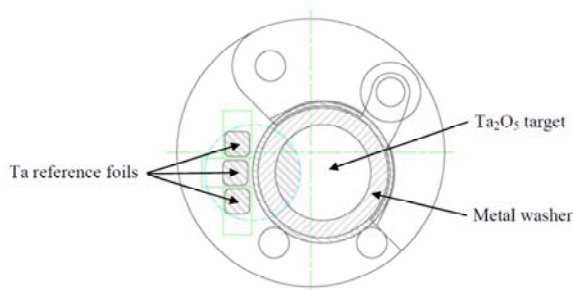


FIGURE 1. Illustration of the test sample in a fixture within the x-ray system. Three calibrated reference foils are used insitu as a baseline for the intensity map generated by the CT system.

To quantify the density of these Tantara samples to the %1.0 level a combination of dimensional measurement, chemical analysis and x-ray radiography techniques have been developed to verify the integrity of the samples before being used in an experiment. The configuration being used for measuring density variation across the sample surface is shown in Figure 1. A set of calibrated reference foils are used in conjunction with the sample under test to compare x-ray transmission intensity while under the same environment in a single set-up. Figure 2 shows an image of the sample foam part. The variation in intensity across the area illustrates density variations.

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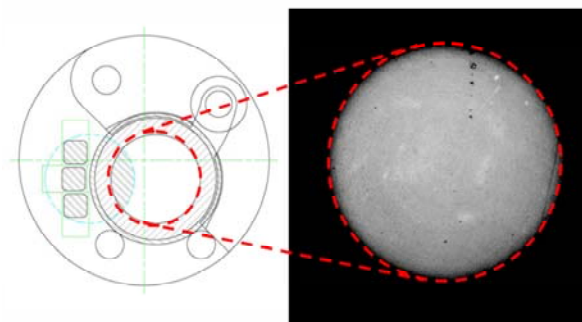


FIGURE 2. Example radiograph showing foam sample. Variations in intensity represent voids or inclusions throughout the bulk of the sample.

The focus of this paper will be to describe the uncertainty and error sources related to determining 500 mg/cc Tantara foam density by combining dimensional metrology, chemical analysis and x-ray radiography.

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Geometric Error Simulation and Evaluation for Machine Tools

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ABSTRACT

In this paper the geometric modeling result of a machine tool and experimental measurement result for R-Test are presented. The geometric modeling has been performed by HTM (Homogeneous-Transformation-Matrix) formulation method from the geometric parameters between machine tool axes. From the HTM formulation those measurement error outputs can be obtained with assumed geometric errors (the forward direction calculation) and the backward direction calculation can be also achieved by the method of Least-Squares-Method which means that those machine's geometric errors can be calculated from the experimental error output data. The R-Test simulation result and identified geometric error result from experimental data are presented.

INTRODUCTION

These days a machining center structure becomes more and more complex in order to enhance flexibility and machining speed. The machine structure is composed of a few motion components that include feeding and rotational tables. According to the configuration of rotational and feeding axes assembled, 5-axis machining centers show various structures and even very flexible structure such as PKM (Parallel Kinematic Machine) which do not have well-defined linear and rotational axes. In these multi-axis machining centers traditional measurement method, which is measuring geometric and motional errors for each feeding and rotational components, generally shows difficulty in measuring geometric errors between motional axes especially when a rotational axis is involved.

In general, 5-axis machining center has many location (geometric) errors and component errors (such as the positioning error and straightness error of each linear axis) and because of the complex structure and assembly, the error measurement method is not yet well

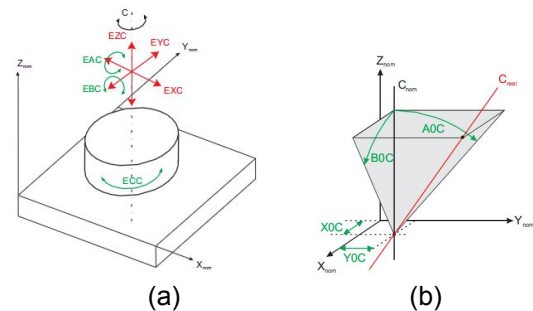


FIGURE 1. Definitions in ISO230-7 : (a) axis average line and error motion of a rotary axis (b) rotary axis involved geometric error

established. The R-test method was proposed for measuring geometric errors of rotary table and it can measure directly three-Cartesian-axis error with submicron measurement resolution. However, the standard general measurement and analysis method has not been yet well established. The R-Test equipment has been designed in our laboratory with three capacitive position sensors. And a datum ball has been used for measuring the relative error displacement.

GEOMETRIC MODELING AND EXPERIMENTAL RESULTS

In order to implement least-squares-method the geometrical modeling of measurement process is required. The HTM (Homogeneous Transformation Matrix) method is used for formulation and the coordinate transformation is made from the datum ball coordinate to the translational table coordinate (to the Tool Center Point) just as seen in the Fig. 2. Basically, the HTM matrix from the reference frame to the 1st coordinate system is to represent the axis offset value. The next HTM matrix which has direction vectors is for the axis orientation and the last HTM matrix is for the rotation of rotary axis table. By multiplying these HTM matrices the mathematical formulation, which describes the motion of the datum ball in the TCP coordinate, can be obtained as shown in the Eq. (2).

$$\begin{aligned}
 {}^R T_1 &= \begin{bmatrix} 1 & 0 & 0 & x_0 \\ 0 & 1 & 0 & y_0 \\ 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 1 \end{bmatrix} \\
 {}^1 T_2 &= \begin{bmatrix} | & | & | & 0 \\ n_1 & n_2 & n_3 & 0 \\ | & | & | & 0 \\ 0 & 0 & 0 & 1 \end{bmatrix} \\
 {}^2 T_3 &= \begin{bmatrix} \cos \theta & -\sin \theta & 0 & 0 \\ \sin \theta & \cos \theta & 0 & 0 \\ 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 1 \end{bmatrix} \\
 q^R &= {}^R T_1 {}^1 T_2 {}^2 T_3 q^3
 \end{aligned} \quad (1)$$

A 3D probe system has been designed for R-test and it is shown in Fig. 3. It uses three capacitive sensors for displacement measurement between a datum ball and the measuring device. Those sensors are located in a manner that each center line passes through center point of the datum ball. The datum ball diameter is 25 mm and has < 50 nm roundness error. The device body material is aluminum which was heat treated and the capacitive sensor is mounted to it by means of screw-lock at the end of sensor.

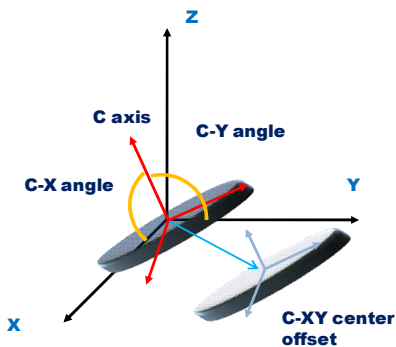


FIGURE 2. Geometric relation between rotary and linear axes for modeling measurement process

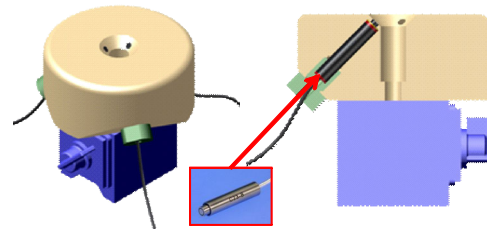


FIGURE 3. Designed 3D probe system schematic showing capacitive sensors

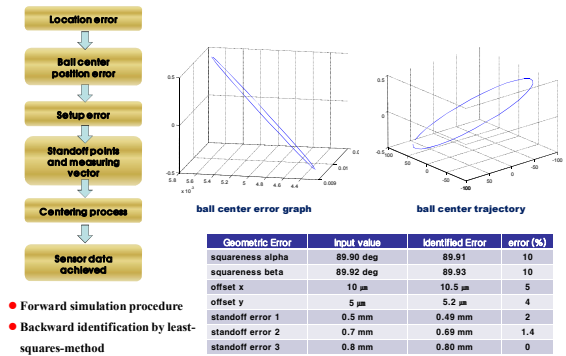


FIGURE 4. The R-Test simulation and error identification

Figure 4 shows the result of the forward and backward simulation. Including the measurement device' setup error and geometric errors, the forward simulation calculates a datum ball center trajectory and by Least-Squares-Method the backward error identification has been achieved. As seen in the table, the identified error is below 10 % for each unknown error.

The R-test experiment has been performed with our lab-built measuring device for a 4-axis ultraprecision machine tool. Those measuring errors are related with the C-axis rotary table and therefore those X and Y axes are controlled simultaneously with the C-axis table to follow a circular trajectory. Those experimental and backward identified results are shown in Fig. 5. With the raw experimental data from the measuring device, x, y and z directional error values can be obtained through a calculation. With the converted x, y, and z directional error data backward identification has been performed and those identified angular and positional error values are shown. The angle alpha means the angle error between the C-axis and X-axis and the angle beta is the error between the C-axis and Y-axis. The offset error means that the

center positional error of the C-axis from the origin of the XY-axis.

CONCLUSION

For further works extracting algorithms for more machine errors will be investigated with those other measuring methods such as laser diagonal method and table rotational error data.

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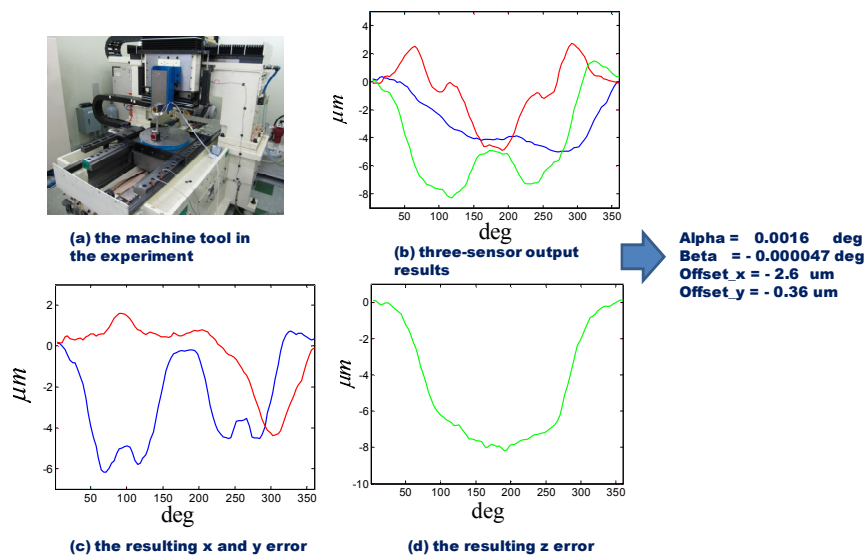


FIGURE 5. (a) the experimental setup is shown. (b) directly measured three-sensor outputs. (c) and (d) converted x, y, z errors. blue : x, red : y, green : z

The Effect of Tip Size in Calibration of Surface Roughness Specimens with Rectangular Profiles

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ABSTRACT

When measuring rectangular and trapezoidal profile roughness specimens, the stylus tip increases the measured profile peak width and decreases the measured valley width. This can cause either an increase or a decrease in the apparent roughness average R_a , depending on the tip size and the ratio of peak width to valley width. Sometimes the change is larger than the combined measurement uncertainty from other sources. This raises the question as to whether measured surface parameters should be corrected for the effect of tip size.

KEYWORDS: Surface metrology, roughness calibration, rectangular profile, stylus radius, Type C roughness specimen.

1. INTRODUCTION

Periodic profile roughness specimens are defined as Type C roughness specimens in the ASME B46-2009 [1] and ISO 5436-1:2000 standards [2] for the calibration of stylus instruments. Examples are specimens with triangular, sinusoidal, arcuate, rectangular or trapezoidal profiles. It is well known that the size and shape of the stylus tip affect measured surface geometry and roughness parameter values. For the measurement of engineering surfaces with fine surface texture, increasing tip size may decrease the measured R_a value because the larger tip does not contact the bottom of narrow valleys. However, for the calibration of Type C roughness specimens with wide profile bottoms, such as specimens with arcuate, rectangular or trapezoidal profiles, the main effect of tip size is to increase the measured peak width and decrease the measured valley width. This may have a significant, and at times counter-intuitive, effect on the measured R_a value. For example, it was observed that an arcuate profile roughness specimen measured with a $0.4\ \mu\text{m}$ tip radius showed an R_a value of $1.260\ \mu\text{m}$ (Fig. 1a). When the tip radius was increased to $5\ \mu\text{m}$, however, the R_a value did not decrease, but

rather increased to $1.332\ \mu\text{m}$ (Fig. 1b)) [3]. The uncertainty of both measurements was estimated to be less than 1.5 % R_a . The R_a difference ($0.072\ \mu\text{m}$, or 5.7 % R_a) was almost 2.7 times as large as the combined measurement uncertainty arising from other sources.

2. CALIBRATION OF RECTANGULAR AND TRAPEZOIDAL PROFILE SPECIMENS

The same effect may occur when measuring rectangular and trapezoidal profile roughness specimens, which are among the most widely used Type C specimens for the calibration of stylus instruments. For a given rectangular or trapezoidal profile with amplitude A and

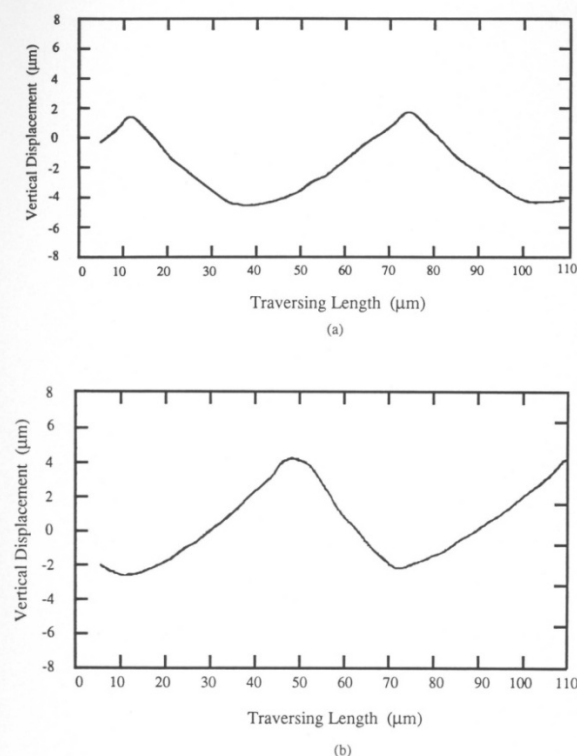


Figure 1. The peak widths of an arcuate profile specimen are enlarged by increasing the stylus size from $0.4\ \mu\text{m}$ (a) to $5\ \mu\text{m}$ (b). The R_a value increased from $1.260\ \mu\text{m}$ (a) to $1.332\ \mu\text{m}$ (b).

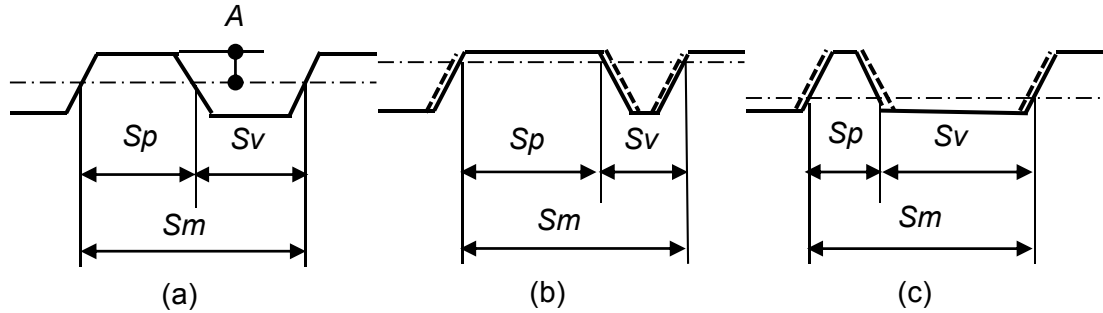


Figure 2. For a trapezoidal profile specimen with amplitude A and wavelength S_m (Fig. 2a), the maximum roughness R_a value occurs when the peak width S_p equals the valley width S_v , or $S_p/S_m = 0.5$ (see Fig. 2a).

wavelength S_m (Fig. 2a), the maximum R_a value occurs when the peak width S_p equals the valley width S_v . For a rectangular profile, the respective R_a equals the profile amplitude A : $R_a(\max) = A$. Either wider ($S_p > S_v$, Fig. 2b) or narrower ($S_p < S_v$, Fig. 2c) peaks result in a lower R_a value. The difference depends on the ratio of S_p/S_m . The R_a decreases if this ratio increases or decreases relative to 0.5.

For the calibration of rectangular and trapezoidal profile roughness specimens, the measurement error in R_a depends on the tip size and profile shape. If the profile peak width is larger than the valley width ($S_p/S_m > 0.5$, Fig. 2b), the tip size increases the S_p/S_m ratio further (see the dashed lines in Fig. 2b), which decreases the measured R_a value. On the other hand, if the peak width is significantly smaller than the valley width ($S_p/S_m < 0.5$, Fig. 2c), the tip size moves the ratio S_p/S_m towards 0.5 (see the dashed lines in Fig. 2c), which increases the measured R_a value. When the peak and valley width of the calibrated specimen are significantly different, the R_a offset caused by the tip size can be significant.

3. TIP SIZE EFFECT

Figure 3 shows a simplified scheme for the estimation of the tip size effect for a rectangular profile specimen. The solid line shows a specimen with peak width S_p and valley width S_v . For simplicity, we assume a cylindrical stylus with radius r and a flat end form, so that the corner rounding and inclined side wall effects caused by a rounded, cone-shaped stylus can be ignored. Because of the tip radius r , the measured peak width is increased to $S_p' = S_p + 2r$, and the valley width S_v is decreased to $S_v' = S_v - 2r$. Assuming a mathematical mean line can be used instead of the least square

fitted profile mean line, the R_a value can be calculated by moving the profile mean line up or down so that the areas above and below the mean line are equal.

$$P_v = \frac{P_t (S_p + 2r)}{S_m}, \quad (1)$$

$$P_p = \frac{P_t (S_v - 2r)}{S_m}, \quad (2)$$

$$R_a = \frac{P_p (S_p + 2r) + P_v (S_v - 2r)}{S_m} \\ = 2P_t(1 - \alpha)\alpha + \frac{4P_t}{S_m}(1 - 2\alpha)r - \frac{8P_t}{S_m^2}r^2 \quad (3)$$

Where P_t is profile height, $P_t = P_p + P_v$, and $\alpha = S_p/S_m$. The last two terms in the expression for R_a represent the errors introduced by the probe radius r .

Based on Eq. 1 to 3, Fig. 4 shows the calculated results for the R_a values of a rectangular profile specimen with a peak width ratio S_p/S_m changing from 0.125 to 0.875, assuming the profile height P_t is 2 μm , the profile period S_m is

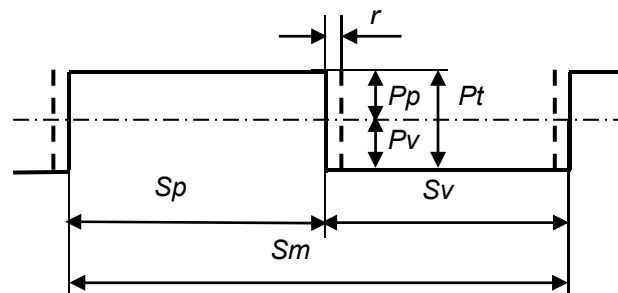


Figure 3. Tip size effect on a rectangular profile specimen.

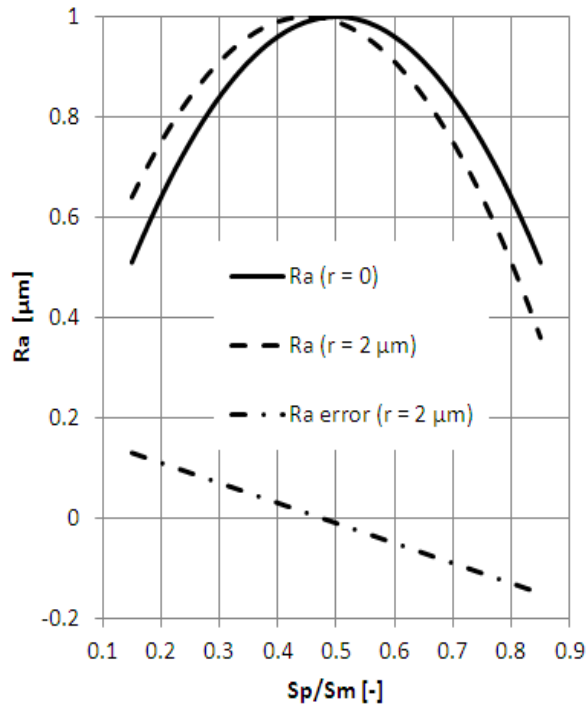


Figure 4. The change of R_a values of a rectangular profile specimen with the peak width ratio Sp/Sm changing from 0.125 to 0.875, assuming the profile height P_t is 2 μm , profile period Sm is 80 μm and the tip radius r is 2 μm .

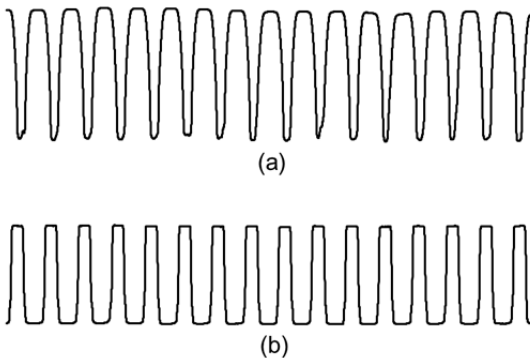


Figure 5: Two rectangular profile roughness specimens having wider peak widths (a), and wider valley widths (b).

80 μm , and the radius r of the cylindrical tip is 2 μm . It can be seen that when the peak width ratio Sp/Sm is equal to 0.45, or $Sp = Sm/2 - 2r = 36 \mu\text{m}$, R_a achieves the maximum value of 1 μm ($0.5 \cdot P_t$). Either a decrease or increase of the profile peak width Sp from that point will cause a decrease of the R_a value. This effect becomes more significant when the profile peak width Sp is extremely wide or narrow.

4 MEASUREMENT RESULTS

Rectangular profile roughness specimens were measured to support the numerical analysis. Two measured specimens are highlighted here [4]: one with larger peak widths than valley widths (Fig. 5a) and the other with larger valley widths than peak widths (Fig. 5b). The measured profiles include the effect of dilation by the stylus tip (nominally a conical tip with 2 μm radius and 45° apex angle). The best estimate of the actual (real) surface profile is obtained by eroding the tip shape from the measured profile using morphological filters [5, 6]. In both cases, there were significant differences in R_a between the measured profiles and the reconstructed profiles. For the surface with wider peaks than valleys, the R_a of the measured profile is smaller than the R_a of the eroded profile (i.e., a negative measurement error). For the surface with wider valleys than peaks, the R_a of the measured profile is larger than the R_a of the eroded profile (i.e., a positive measurement error). In both cases, the relative error in R_a is more than 1.5 %, exceeding the combined expanded uncertainty from other sources. Changes in measured R_a are significant even for a modest change in tip radius—for example, from 2 μm to 1.5 μm . Detailed measurements and simulations can be found in Ref. 4.

5. DISCUSSION

The effect of tip radius on R_a raises the question as to whether the measured surface parameters should be corrected for the tip size effect. According to the GUM [7], correction is required for any significant systematic effects in measurement results:

It is assumed that the result of measurement has been corrected for all recognized significant systematic effects and that every effort has been made to identify such effects. [7]

The answer to the question depends on the definition of the measurand. Correction is not required if the measurand is the R_a of the profile obtained using a tip with a stated radius. However, if the measurand is the R_a of the actual or mechanical surface, correction may be required. In the ISO 25178-2:2012 standard [8], the “mechanical surface” is defined as

Boundary of the erosion, by a spherical ball of radius r , of the locus of the center of an

ideal tactile sphere, also with radius r , rolled over the skin model of a workpiece. [8]

This implies the use of correction for significant systematic effects caused by the tip size.

Many laboratories likely do not use tip size correction for reporting surface roughness measurements. In some cases, it may be a major contributor to measurement differences. In order to achieve measurement agreement in the calibration of rectangular and trapezoidal profile roughness specimens using different tip sizes, the authors suggest that tip size corrections should be performed.

The tip size effect is particularly important for the measurements of smooth engineering surfaces with Ra less than $0.1\text{ }\mu\text{m}$. Significant differences were found in surface measurement comparisons using random profile roughness specimens with Ra ranging from $0.01\text{ }\mu\text{m}$ to $0.1\text{ }\mu\text{m}$ [9, 10]. It was suggested that the effect of tip size differences were largely responsible for the observed differences in Ra [3].

The quality control of smooth engineering surfaces ($Ra < 0.1\text{ }\mu\text{m}$) becomes increasingly important, not only because of their important engineering functions, but also their high production costs. NIST aims to develop high-precision, random-profile roughness specimens as a Standard Reference Material (SRM) to support smooth surface measurements in U.S. industry. Such a project could include the development of a tip size correction program for the measurements of smooth engineering surfaces with random profiles.

6. SUMMARY

For the calibration of Type C roughness specimens with wide profile bottoms, such as specimens with arcuate, rectangular or trapezoidal profiles, the main effect of tip size is the increase of peak width and decrease of valley width. This may have a significant, and at times counter-intuitive, effect on the measured Ra value. The resulting systematic offset can be larger than the reported measurement uncertainty. In order to achieve measurement agreement when using different tip sizes, the authors of this work suggest that a correction for tip size be performed.

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A NEW SUCCESSIVE THREE POINT METHOD OF ROUNDNESS MEASUREMENTS

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INTRODUCTION

The successive three point method is an useful technique for on-machine measurement of roundness especially at the factory site because of its simple and easy calculation algorithm. However the method has a restriction in detectable spatial frequency. Supposing that the spacing angles between probes are $2\pi/N$, harmonic components the order of which are equal to the integral multiple of N cannot be detected. Where, N is an integer dividing the whole circumference of the specimen into N equal parts. The harmonic components higher than $N/2$ order also cannot be detected. In this study, a new successive three point method is proposed without undetectable spatial frequency component using recurrence formula and the effectiveness of the method is confirmed by numerical simulation.

PRINCIPLE OF THREE POINT METHOD FOR ROUNDNESS

The arrangement of three displacement sensors is shown in Fig.1. $f(\theta)$ is radial figure of the specimen, $m(\theta)$ is output of the sensor respectively and φ is the spacing angles between probes. $e_x(\theta)$ and $e_y(\theta)$ are the error motion of the rotary table. The discontinuous radial figure data are obtained at equal angular

interval of φ along a circumference line of the specimen as shown in Fig.2. When 2π is not divisible by φ without remainder, the radial figure profile can be evaluated without any undetectable harmonic component. The periodic revolution M , which is the number of revolution necessary for the interval to obtain data at an identical position along the circumference, is larger than 2.

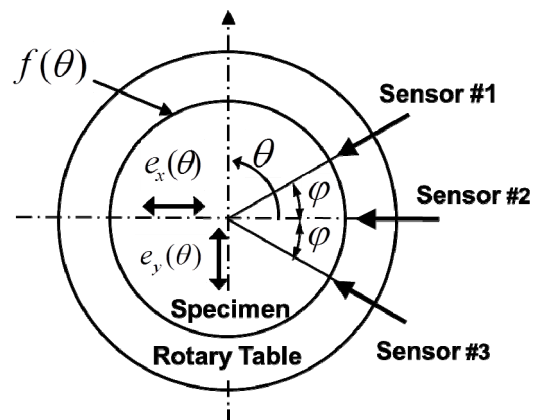


Figure 1. Arrangement of sensors in successive three point method for roundness measurement

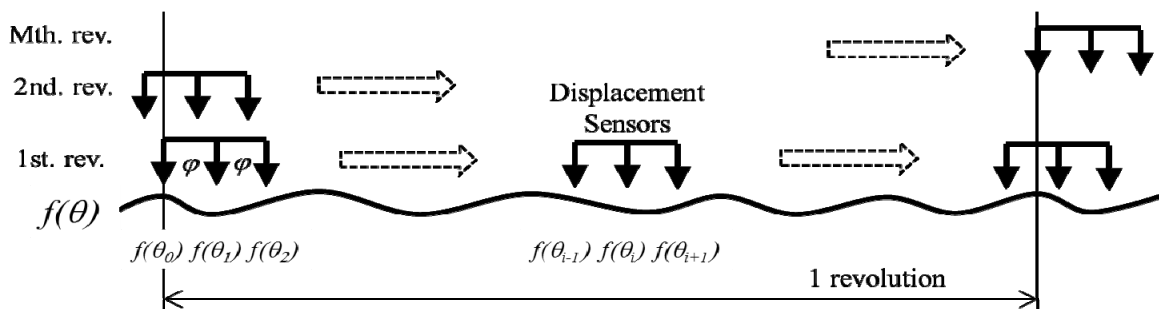


Figure 2. A concept of roundness measurement by successive three point method

The number of data point n within the period can be expressed as Eq. (1).

$$n = 2\pi * M / \varphi \quad (1)$$

The output signals of the sensors are expressed in Eqs.(2) to (4).

$$m_1(\theta_i) = f(\theta_{i-1}) + e_x(\theta_i) \cos \varphi - e_y(\theta_i) \sin \varphi \quad (2)$$

$$m_2(\theta_i) = f(\theta_i) + e_x(\theta_i) \quad (3)$$

$$m_3(\theta_i) = f(\theta_{i+1}) + e_x(\theta_i) \cos \varphi + e_y(\theta_i) \sin \varphi \quad (4)$$

To eliminate the error motion $e_x(\theta_i)$ and $e_y(\theta_i)$, the differential output signals are calculated using Eqs.(5) and (6).

$$\mu_1(\theta_i) = m_1(\theta_i) - m_2(\theta_i) \cos \varphi \quad (5)$$

$$\mu_2(\theta_i) = m_2(\theta_i) \cos \varphi - m_3(\theta_i) \quad (6)$$

The second difference of the output signals $\Delta\mu(\theta_i)$ is expressed by Eq.(7).

$$\begin{aligned} \Delta\mu(\theta_i) &= \mu_1(\theta_i) - \mu_2(\theta_i) \\ &= f(\theta_{i-\varphi}) - 2f(\theta_i) \cos \varphi + f(\theta_{i+\varphi}) \end{aligned} \quad (7)$$

The recurrence formula of the radial figure profile is obtained as Eq.(8).

$$f(\theta_{i+\varphi}) = \Delta\mu(\theta_i) + 2f(\theta_i) \cos \varphi - f(\theta_{i-\varphi}) \quad (8)$$

Eq.(8) shows that the radial figure $f(\theta_{i+\varphi})$ can be calculated using $f(\theta_i)$ and $f(\theta_{i-\varphi})$ with the second difference of output signals of the sensors #1 to #3. To solve successively the recurrence formula, two initial values of radial figure, $f(\theta_0)$ and $f(\theta_1)$, when the sensors #1 and #2 are at the initial start position, $\theta = -\varphi$, respectively.

INFLUENCE OF ERROR IN INITIAL VALUE $f(\theta_0)$ AND $f(\theta_1)$ ON MEASURED ROUNDNESS

The influence of the error included in the initial value is investigated. If the section of specimen is a true round and the arrangement of sensors is geometrically perfect, both of the initial values $f(\theta_0)$ and $f(\theta_1)$ can be considered to be 0. However, in case of the initial value of $f(\theta_1)$ is not 0, due to a kind of setting error, a 1st harmonic component appears in the figure profile. Fig.3 shows the calculated profile error by the recurrence formula when $f(\theta_1)$ is assumed to be 1% of the full measuring range of the displacement sensor used in the apparatus. The intensity of figure error is normalized by the full measuring range of the displacement sensor. Where, the angle φ between probes are supposed to be $\pi/4$.

The 1st harmonic component is appeared as a result of a setting error between the center of specimen and the intersection of three probe axes as shown in Fig.4. The 1st harmonic component can be removed by a harmonic analysis of measured radial figure profile using FFT analyzer.

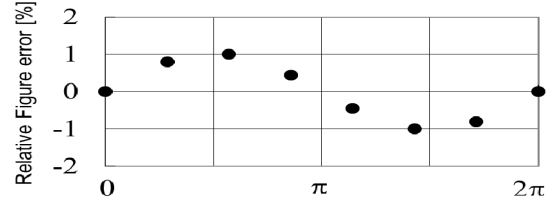


Figure 3. 1st harmonic component produced by an error in initial value of $f(\theta_1)$

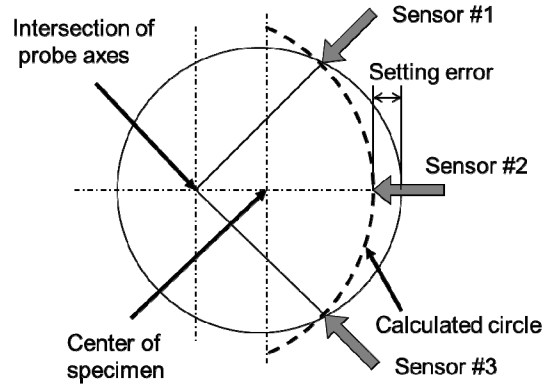


Figure 4. Setting error of sensors causing an error in initial value $f(\theta_1)$

EVALUATION OF EFFECTIVENESS OF THE METHOD PROPOSED BY SIMULATION

To confirm the effectiveness of the method proposed, three types of numerical simulations are carried out.

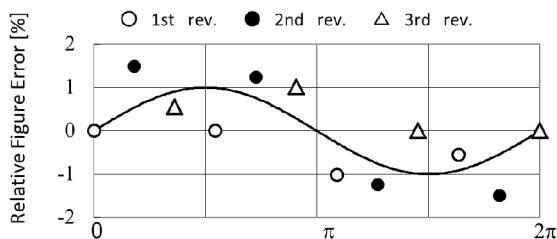
(1) Specimen with 3rd harmonic component

Fig.5(a) shows the result of the simulation measuring the specimen including 3rd harmonic component the relative amplitudes of which are 1%. The spacing angles φ between probes and the initial value $f(\theta_1)$ are supposed to be $6\pi/11$ and 0%, respectively. An inevitable error included in $f(\theta_1)$ causes a 1st harmonic component as shown in Fig.5(a) by a solid line. Figure 5(b) shows the figure profile after removing the 1st harmonic component from the measured data shown in Fig.5(a). Fig.5(c) is the roundness profile shown in Fig.5(b) expressed by polar coordinates. The 3rd harmonic component can be clearly observed. These results suggest the effectiveness of new method proposed.

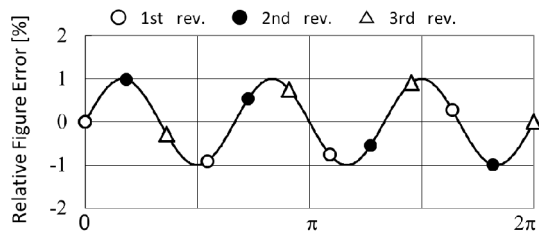
(2) Specimen with 3rd and 5th harmonic components

Fig.6(a) shows the result of simulation measuring the specimen including 3rd and 5th harmonic components the amplitudes of which are 0.5% and 1.0%, respectively. The

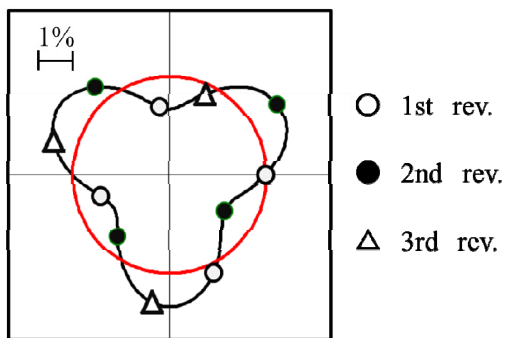
spacing angle between probes φ and the initial value $f(\theta_i)$ are supposed to be $6\pi/11$ and 0, respectively. The 1st harmonic component caused by the setting error is shown in Fig.6(a) by solid line. Fig.6(b) shows the figure profile after removing the 1st harmonic component from the measured data shown in Fig.6(a). Fig.6(b) shows the result of roundness profile shown in Fig.6(c) expressed by polar coordinates. Both of the 3rd and 5th harmonic components can be clearly observed.



(a) Figure profile including a 1st harmonic component

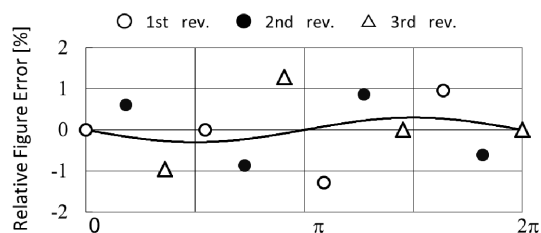


(b) Figure profile after removing of 1st harmonic component

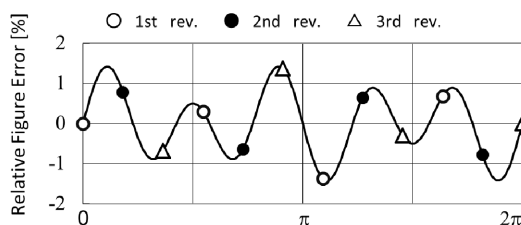


(c) Roundness expressed by polar coordinates

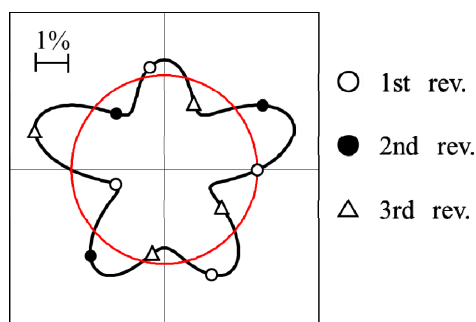
Figure 5. Result of simulation measuring the specimen including 3rd harmonic component.



(a) Figure profile including a 1st harmonic component



(b) Figure profile including a 1st harmonic component

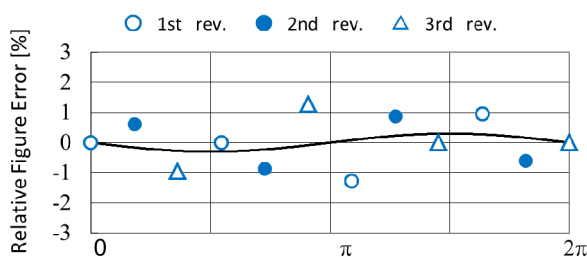


(c) Roundness expressed by polar coordinates

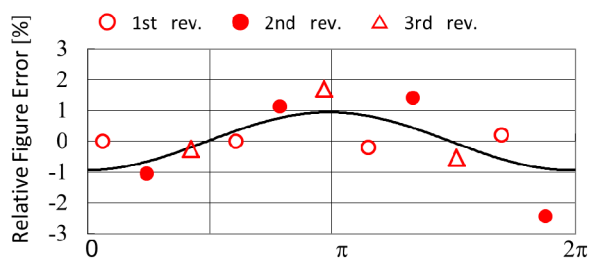
Figure 6. Result of simulation measuring the specimen including 3rd and 5th harmonic components.

(3) Combination with an interpolation method
To obtain more detailed figure profile by increasing the number of data points, an interpolation method was proposed by the authors in ASPE 2008 annual meeting. In the method, some series of data obtained by shifting the start point $f(\theta_i)$, are superimposed. Fig. 7 shows the result of simulation measuring the same specimen as that used in Figure 5. Fig.7(a), (b), and (c) are three different series of figure data obtained by shifting the start point by 0, $2\pi/33$ and $4\pi/33$, respectively. These figure profiles individually include different 1st harmonic components. The superimposed figure

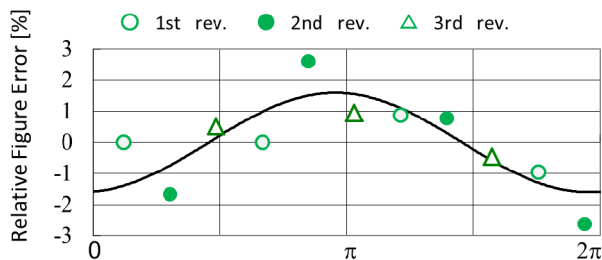
profile after removing the individual 1st harmonic component from each figures is shown in Fig.7(d) and (e).



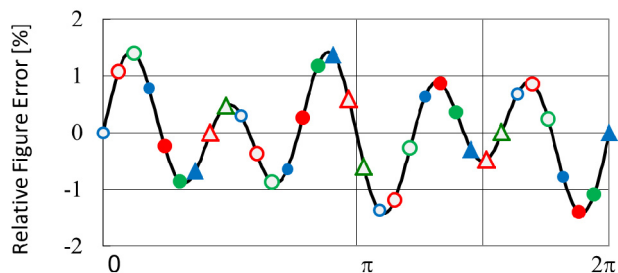
(a) Starting angle $\theta_0 = 0$



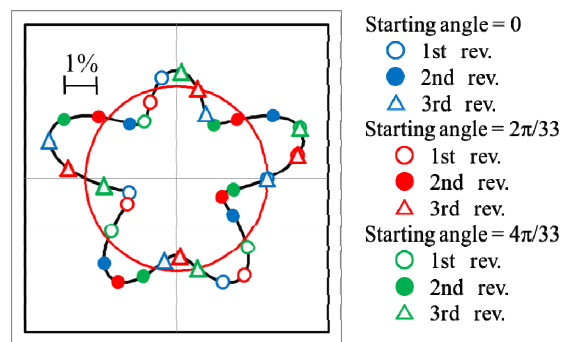
(b) Starting angle $\theta_0 = 2\pi/33$



(c) Starting angle $\theta_0 = 4\pi/33$



(d) Superimposed figure profile after removing of 1st harmonic component



(e) Roundness expressed by polar coordinates

Figure 7. Result of simulation measuring the specimen including 3rd and 5th harmonic components using an interpolation method.

SUMMARY

A new successive three point method is proposed for roundness measurement without undetectable harmonic component. The effectiveness of the method is confirmed by numerical simulation.

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Authors Index

Abbas, Ali E. 50
 Abe, Takeyuki 296
 Abolbashari, Mehrdad 256
 Afoakwa, Richard 464
 Aguirre, Gorka 376
 Ahmar, Elie 410
 Al-Bender, Farid 182
 Alger, Ethan T. 320
 Allen, James 42
 Al-Qhatani, Hussain 518
 Althagafy, Khalid T. 457
 Anderson, Hyrum 252
 Arneson, David A. 30, 180, 206, 217, 300,
 321, 410, 461
 Barker, D. 320
 Bauza, Marcin B. 20, 363
 Beaucamp, Anthony Tadeus 414
 Benjaminson, Emma Claire 402
 Bennett, Donald W. 380
 Berg, Trenton 66
 Bettencourt, Kerry 329
 Bhandarkar, Suhas 320
 Blobaum, Kerri J. M. 355
 Bos, Edwin 85
 Boye, Robert 430
 Bradford, John W. 224
 Brandenburg, Nolan 464
 Brecher, Christian 410
 Buechner, Hans-Joachim 103
 Bui, Chinh Ba 367, 492
 Buice, Eric S. 380
 Burnham-Fay, Ethan 288
 Buruaga, Miren 376
 Campbell, Ronald B. 402
 Cappa, Steven 182
 Castro, Carlos 371
 Cavanaugh, Dennis 252
 Chao, Leon Shih 464
 Chen, Liang-Chia 468
 Chen, Shih-Chi J. 186
 Chesna, Jacob W. 472
 Chetwynd, Derek G. 457
 Cheung, Chi-Fai C.F. 418, 496
 Cho, Hyun-Gwang 190, 394

Choi, Byung Joo 272
 Choi, Doo-Sun 433
 Choi, Hwan-Jin 433
 Chon, Kwon Su 260
 Chu, Yong S. 116
 Chung Ph. D., Sung-Chong Joe . . 190, 220, 382, 390
 Cilip, Christopher M. 363
 Clark, Daniel E. 73
 Coupland, Jeremy M. 15
 Culpepper, Martin Luther 54, 186, 406
 Dalrymple, Timothy M. 476
 Davies, Angela D. 256, 488
 Davies, Matthew A. 500
 de la Maza, Borja 303
 Deakyne, Theodore Roy 206, 209, 422
 DeBra, Daniel B. 73
 Del Orbe, Dionisio Valentin 304
 Devitt, Andrew J. 42, 245
 Donmez, M. Alkan 484
 Dow, Thomas A. 62, 107, 124, 128, 325
 Drabarek, Pawel 70
 Du, Lucy 54
 Dumitrescu, Eugen 284
 Dutterer, Brian S. 339
 Dzenitis, Elizabeth G. 320
 Eberhardt, Ramona 386
 Edson, Stanley L. 194, 320
 El-Dasher, Bassem 355
 ElKott, Diaa F. 480
 Ellis, Jonathan D. 77, 280, 288
 Erez, Shmuel 244
 Erkorkmaz, Kaan 140, 160, 173, 236
 Evans, Christopher J. 488
 Farahi, Faramarz 256
 Felker, Sean 329
 Fellows, Joseph 252
 Fesperman Jr., Ronnie R. 484
 Fitsos, Pete 194
 Fleischer, Matthias 70
 Florando, Jeffery N. 355
 Foreman, Matthew 15
 Forest, Craig R. 152
 Fucetola, Jay J. 245, 292
 Fujii, Katsuji 426

Furuki, Tatsuya	426, 449
Furushiro, Naomichi	308
Furutani, Katsushi	198
Ganguly, Vasishta).	58, 264
Garrard, Kenneth P.	62, 325
Garrison, Stephen James	132
Geraghty, Paul	202
Gerges, Awad S.	256
Ghantasala, Muralidhar.	304
Ghazaly, Mariam Md	144
Gillmer, Steven	77, 288
Giusca, Claudiu	15
Glöss, Rainer.	99
Gomez-Acedo, Eneko	312
Gonzales, S.	320
Gorostiaga, Mikel	376
Gossel, Marcus.	386
Graber, Tim	30
Graziano, Arthur	58
Greenfield, Nathan	66, 276
Gross, Richard J.	355
Guan, Jun	268
Hall, James	194
Hamaguchi, Tetsuya	316, 343, 359
Hanson, Bruce	300
He, Bo-Ching.	468
He, Liangyu	488
Heberley, Brian	398
Heidl, Brian F.	320
Heidler, Nils	386
Heilmann, Ralf K.	245, 292
Heinbockel, C. L.	371
Heinisch, Josef	284
Henning, Robert	30
Her, Ting-Hua	472
Hesse, Steffen	103
Higashijima, Genki	169
Hiroi, Tetsumaro	240
Ho, Lai-Ting	418
Hocken, Robert J.	111, 510
Holst, Gregory L.	152
Hopkins, Jonathan B.	26
Horner, Jeffrey B.	320
Hosseinkhani, Yasin Y.	160
Howland, Fred A.	320
Hwang, Jooho.	367, 492, 527

Ikeshima, Shota	316, 359
Ismail, Fathy	173
Jacobs, Ben	514
Jared, Bradley H.	430
Je, Tae-Jin	433
Jeon, Eun-Cheae.	433
Jeon, Yongho	272
Jeong, Jaehwa	272
Jordan, Scott C.	164
Kagohashi, Yusuke	449
Karandikar, Jaydeep M.	50
Katagiri, Masato	296
Kato, Daisuke	169
Kato, Tomonori	169
Katzschmann, Michael	99, 103
Keller, Eric E.	206, 209, 422
Khim, Gyungho	213
Kim, Byung-Sub	437
Kim, Chang-Ju.	148
Kim, Gyungho	382
Kim, Jongsoo	148
Kim, Jungdae	116
Kim, Myungjune	390
Kim, Taehyun.	272
Kim, Young-Pyo.	437
Kimura, Atsushi.	331
Kiyono, Satoshi	534
Knapp, Byron R.	30, 206, 209, 422
Knebel, Richard.	480
Ko, Tae Jo	335
Kobayashi, Yoshikazu	347
Köchert, Paul.	268
Koh, Kyun Hee	422
Kong, Ling-Bao L.B.	418, 496
Kortaberria, Gorka.	312
Kozicki, David E.	66, 276
Kroll, Jeremy J.	320
Kumar, Chockalingam.	329
Lai, Barry	116
Lane, Brandon M.	128
Langehanenberg, Patrik	284
Lantz, Brian T.	73
Larson, Mark W.	132
Lauer, Kenneth	116
Lawson, Benjamin L.	371
Leach, Richard K.	15

Lee, Chan-Hong	492, 527
Lee, Moon-Gu	272
Lee, Sung Cheul	372
Lettenbauer, Hubert	20
Lewinsohn, Charles	252
Liang, Rong	99
Liebers, Melvin J.	217, 321, 461
Lineberger, Jennifer Lynn	500
Liou, Huay-Chung	468
Liu, Wenjun	116
Lord, Dawn	329
Lu, Xiaodong	81
Lüerß, Bernd	284
Mandal, Rahul	15
Marsh, Eric R.	30, 206, 209, 422
Marston, Zachary C.	325
Maser, Jorg	116
Matsumori, Noboru	308
Matsumoto, Akihiro	414
McCalib Jr., David B.	26
McKinley, Jared Dell	304
Medicus, Kate M.	504
Miller, Glenn J.	38
Min, Bok-Ki	190, 220, 390, 394
Mirkarimi, Paul	329
Mizoguchi, Tomohiro	347
Mochida, Hideki	224
Mock, Christopher Daniel	99
Moers, Ton	85
Mohr, Hans-Ulrich	99
Montesanti, Richard C.	380
Moylan, Shawn P.	484
Muralikrishnan, Bala	530
Myers, Matthew D.	371
Nagato, Keisuke	46, 316, 343, 359
Nakao, Masayuki	46, 316, 343, 359
Namba, Yoshiharu Y.	260, 414
Nathan, B.	320
Nazaretski, Evgueni	116
Nederbragt, Walter W.	329
Nguyen, Cao Ngoc	367
Nguyen, Minh Hop M.	506
Ni, Kang	510
Niglas, Paul	173
Nishibashi, Ken	343
Nowak, W. John	107

Nowakowski Ph.D., Bartosz K.	228
Nozaki, Yusuke	169
Nyovanie, Prosper.	54
Ogura, Hidekazu	308
Oh, Jeong-Seok	213
Ohya, Jun	331
Okada, Akira	331
Okada, Mutsumi	426, 449, 453
Okamoto, Yasuhiro	331
Okwudire, Chinedum Emmanuel.	148
Olarra, Aitor	312
Ono, Manabu	232, 240
Ono, Masataka	198
Ono, Takenori	441
Oss, Daniel D.	30
Osumi, Nobuyuki	136
Otsubo, Tatsuki	169
Packeiser, Joerg	363
Pak, Nikita	152, 390
Panas, Robert Matthew	54, 186
Park, Chun-Hong	213, 367, 382, 390, 394
Park, Eun-Suk	433
Park, Jong-Kweon	372, 437
Parks, Robert E.	514
Patten, John A.	304
Peaslee, P.	194
Peijnenburg, Antonius T.	249
Peterson, Shawn C.	329
Petrzelka, Joseph	518
Phaneuf, Christopher Russell	152
Pont, Pere C.	303
Pratt, Daniel S.	363
Pratt, Jon R.	464
Ragan, Timothy	186
Raja, Jayaraman	522
Ramasamy, Suresh Kumar	522
Rao, Niankun	81
Ravindra, Deepak	304
Reibel, Randy Ray	66, 276
Renegar, Thomas Brian	530
Reynaerts, Dominiek	182
Reynolds, Jack L.	380
Risse, Stefan	386
Rivers, Ryan I.	472
Ro, Seung-Kook	372
Robichaud, Guillaume	62

Roblee Ph.D, Jeffrey W.	514
Roos, Peter Aaron	66, 276
Rudolf, Christian	99
Rusnak, Brian	194
Ruszkowski, Angelica	236
Saad, Mohamed El-Sayed	111
Saavedra, Michael P.	430
Sain, John	526
Sakamoto, Makoto	232, 240
Samara, Ayman M.	472
Sanner, Steven L.	217, 321
Saragih, Agung Shamsuddin	335
Sasahara, Hiroyuki	136, 296, 445
Satcher, Joe H.	526
Sato, Kaiji.	144
Saunders, Daniel C.	152
Scattergood, Ronald O.	107, 124
Schaeffel, Christoph	99, 103
Schattenburg, Mark Lee	245, 292
Schenk, Christoph	386
Schlamming, Stephan	464
Schmitz, Tony L.	50, 58, 120, 264, 339
Seifert, Frank	464
Sellmann, Florian	177
Seugling, Richard M.	355, 526
Shelef, Ben	244
Sheridan, Tim	300
Shikazono, Naoki	46, 343
Shim, Jongyoup	527
Shimada, Shoichi	308, 534
Shimamoto, Kazuki	46
Shimomura, Mitsuru	445
Shin, Dong-Soo	220
Shinagawa, Shunta	343
Shirai, Kenji	347
Shu, Deming	116
Slocum, Alexander H.	398, 402
Smith, Douglas T.	228
Smith, Richard C. G.	77, 280, 288
Smith, Stuart T.	228, 472
Smythe, Robert A.	514
Soemers, Herman M.	34
Soetens, Hans	34
Sohn, Alex.	62, 325
Song, Jun-Feng John	530
Soons, Johannes A.	530

Stickler, Daniel	284
Stover, Erik	514
Stubley, Gordon	236
Sung, Edward	245, 292
Suzuki, Hirofumi Y.	426, 449, 453
Swank, Aaron J.	89
Sweatt, William C.	430
Swift, David W.	329, 355
Takasugi, Akio	136
Tanaka, Katsutoshi	169
Tarbutton, Joshua	77
Taylor, John S.	320
Telleria, Maria Jose	406
Thai, Nguyen Van	468
Thigpen, Richard	202
Thompson, Daniel C.	93
To, Suet Sandy	496
Tutsch, Ranier	268
Tyler, Christopher	120
Uda, Yutaka	534
Urreta, Harkaitz	376
Vallance, Ryan R.	209, 422
van der Straaten, Gerrit M.	249
van Rens, Piet	249
van Riel, Martijn	85
Veldhuis, Stephen C.	480
Ventura, Toni	303
Villarrubia, John S.	530
Vonhof, Scott A.	320
Vorburger, Theodore V.	530
Walenda, Carolin	99
Wang, Albert	371
Wang, Chen	280
Wang, Zhigang	132
Watanabe, Akira	347
Watanabe, Takaya	351
Watanabe, Yasuaki	46
Wegener, Konrad	177, 506
Weichert, Christoph	268
Weikert, Sascha	177, 506
Wenzel, Christian	410
White, Daniel	264
Wilson, Christopher R.	276
Wilson, Curtis S.	156
Wilson, Merrill A.	252

Wilson, Michael John	329, 355
Wong, Anthony R.	398
Woody, Shane C.	77
Yamada, Yohei.	136
Yamaguchi, Hitomi	58
Yamaguchi, Tetsuro.	224
Yamaguchi, Tomomi	308
Yamauchi, Takuma	316, 359
Yamauchi, Toshiyuki	331
Yan, Hanfei	116
Yazawa, Takanori.	169
Yeom, Hyun Ho.	272
Yong, S.	116
Yoo, Hongki.	272
Youden, David H.	93
Young, Peter	526
Yun, Janet Love.	264
Zdanowicz, Erik M.	124
Zheng, Alan X	530
Zirkle, Thomas	464
Zubieta, Mikel	312

